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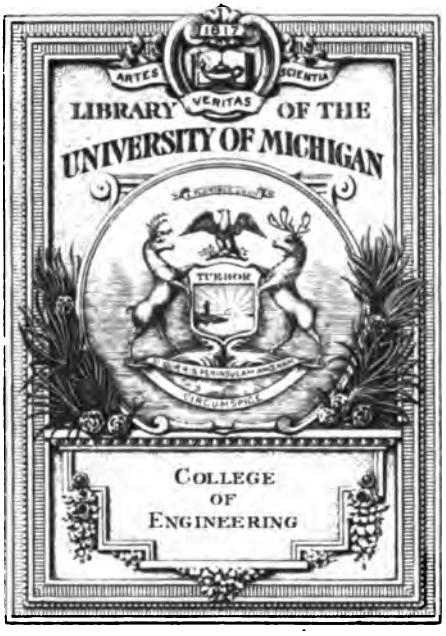
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MARINE ENGINES AND BOILERS

Their Design and Construction

A HANDBOOK FOR THE USE OF STUDENTS, ENGINEERS, AND NAVAL CONSTRUCTORS

BASED ON THE WORK

BERECHNUNG UND KONSTRUKTION DER SCHIFFSMASCHINEN UND -KESSEL

BY

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With over 550 Illustrations and Numerous Tables

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PREFACE TO THE ENGLISH EDITION.

WHEN invited by the Publishers to undertake the editing of the English Edition of Dr Bauer's valuable work on "Marine Engines and Boilers," it was with extreme hesitancy that I contemplated undertaking the work on account of the many heavy calls upon my time in another direction. I was, however, ultimately persuaded to do so on the Publishers securing the collaboration of Messrs E. M. and Sidney Donkin, to whose lot fell the duty of the original rough translation and the conversion of the formulæ into English measures. It would have simplified our labours enormously had we been able to adhere to the original metrical figures without giving their English equivalents. Though most English engineers are conversant with the metrical system, they are, however, still accustomed to think and make their mental comparisons in pounds, feet, and inches; and, as the book is primarily intended for young engineers and draughtsmen, we felt that it would be more useful, and many a mental exercise avoided, if English units were used throughout. I trust, therefore, that the work in its present form may be of value and assistance to the younger generation of marine engineers and naval architects of this country.

As a work of this nature involves an enormous number of conversions, some errors, in spite of the precautions taken to eliminate them, will, we fear, inevitably have crept in. We

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should, therefore, deem it a favour if our readers would call our attention to any cases coming under their notice.

Thanks are due to Mr E. H. Sprague, of University College, London, for checking many of the calculations, to Engineer-Lieutenant H. J. Loveridge, R.N., of the Admiralty, and to Mr Charles Dresser for kindly running through the proofs.

LESLIE S. ROBERTSON.

28 VICTORIA STREET,
WESTMINSTER, S.W.
February 1905.

DR BAUER'S PREFACE TO THE FIRST EDITION.

THE present work owes its origin to an oft-felt want of a condensed treatise, embodying the theoretical and practical rules used in designing marine engines. It contains also drawings of the more usual types of engines, and tables of the various data required in ordinary practice.

The need for such a work has been felt by most engineers engaged in the construction and working of marine engines, by the younger men, as well as by those of greater experience. Probably many of those connected with the building and driving of stationary engines will also welcome the book, as an addition to the literature in which they are more specially interested. To them the chapters on Calculation of Cylinder Dimensions, Turning Moment, Balance of the Moving Parts, Details of the Engine, Piping, Pumps, &c., will be found more particularly useful.

In accordance with the general character of the book, and the necessity of limiting the range of subjects treated, only the most modern types of engines—those of the vertical type used in screw steamers—and of boilers in ordinary use, have been dealt with. Special types have not been dealt with. The greater part of the book is based upon results obtained in actual practice; but little has been taken from other writers on the subject, and then only from specially trustworthy sources.

The compilation of the work and of the drawings has been greatly assisted, or rather rendered possible, by the kind help afforded by a number of the principal firms in Germany and abroad, who have very liberally placed a large quantity of material at the author's disposal. To all of these he desires to

express his grateful thanks, but most of all to the Stettiner Maschinenbau-A.-G. "Vulkan," who have permitted him to make use of a large number of working drawings and photographs.

The work has been compiled as follows:—Mr Ludwig undertook—Part I. Section 5, Sections 49 to 91, and Sections 141 to 153; Part II.; Part IV. Sections 2 to 5; Part V. Sections 2 and 7; while Mr Boettcher undertook — Part I. Section 2; Part V. Section 1; and Part VI. The whole of the remainder was written and compiled by the undersigned, with the constant help of Mr Foettinger.

In judging the book, it must be remembered that it is the fruit of the few hours of available leisure snatched from the midst of a strenuous professional life. Hence it will often be found to lack the necessary finish, and to bear evidence of various shortcomings. It is however hoped that as the book has been written in response to an actual daily felt need, it may, for that very reason, be of the greater service.

In conclusion, the Author would be glad if his readers would bring to his notice any errors and omissions so that he may be able to rectify them should, in the future, a new edition be called for.

Dr G. BAUER,

Engineer to the Vulcan Works, Stettin.

STETTIN, *March* 1902.

DR BAUER'S PREFACE TO THE SECOND EDITION.

As the first edition was exhausted in little more than a year—much sooner than I had ventured to hope—I was unable, much to my regret, to undertake more than a small part of the revisions and alterations which appeared to me to be necessary in the preparation of a second edition. Nevertheless I have found it possible to make good various omissions, and to considerably enlarge the chapters on "Arrangement of the Main Engines" and "Water Tube Boilers," and also to add throughout the book Tables giving data of engines which have been actually constructed. I attach particular importance to the addition of these Tables, because they are of far greater assistance in determining the most suitable dimensions for new engines and machinery, than any number of comparative values or empirical formulæ.

Additional value is given to these Tables because His Excellency the Secretary of State for the Imperial Navy has allowed me to use data compiled from various ships belonging to the Imperial German Navy, and I desire to avail myself of this opportunity of tendering my best thanks to His Excellency.

I desire also to express my indebtedness to the Stettiner Maschinenbau-A.-G. "Vulkan," who have again placed very valuable material at my disposal in order to complete the illustrations of the book.

My thanks are also due to the other firms, in Germany and elsewhere, who have supplied me with material for the second edition of the book, as also to my publishers, who have contributed greatly to the success of the work by their excellent workmanship in the first edition.

In spite of the great, not to say fantastic, expectations entertained in some quarters in regard to the future of steam turbines for ships, I have not ventured at present to add a chapter on the subject to this edition. Although the Parsons turbine has had considerable success *on land*, and has already been fitted in a number of ships employed in running regular services, reliable results from these latter, especially as regards the consumption of coal, are not yet available.

In this second edition, as in the first, Messrs Ludwig and Foettinger have rendered me valuable assistance, while the chapters contributed by Mr Boettcher have not required alteration.

I would esteem it a favour if the reader will bring to my notice any discrepancies he may discover, as was kindly done upon the appearance of the first edition.

Dr G. BAUER,

Engineer-in-Chief of the Vulcan Works, Stettin.

STETTIN, *January* 1904.

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PART I.

THE MAIN ENGINES.

SECTION I.

DETERMINATION OF CYLINDER DIMENSIONS.

§1. **Horse-power.**—The unit employed to measure the rate at which work is done in a steam engine is the "horse-power," *i.e.*, the power exerted in the performance of 33,000 ft. lb. of work per minute.

A distinction is made between what is called *indicated horse-power* (I.H.P.) and the actual or *brake horse-power* (B.H.P.).*

Let $\frac{\pi D^2}{4} = A$ be the area in square inches of a steam piston having a diameter D; let p_m be the mean pressure upon the piston in pounds per square inch † during one revolution; r = radius of the crank in feet; $2r = s$, the stroke; n = number of revolutions per minute; then the power developed in a double-acting cylinder in *indicated horse-power* will be—

$$\text{I.H.P.} = \frac{p_m \times 2s \times A \times n}{33{,}000}$$

and the mean speed of the piston will be—

$$c = 2sn \text{ feet per minute.}$$

Therefore

$$\text{I.H.P.} = \frac{p_m \times c \times A}{33{,}000}$$

Hence by indicated horse-power is understood the work done per minute by the steam on the piston of the engine. Part of this is lost by friction in the working parts of the engine and the shafting. If these frictional losses are deducted from the indicated horse-power, we get the *actual work* available, or the *brake horse-power*, which is the rate at which useful work is done in turning the screw or propeller, or in running the shaft with an artificial brake on.

The brake horse-power in very large engines is rather less, and in small engines considerably less than the indicated horse-power.

If we take B.H.P. $= \eta \times$ I.H.P., the following table (calculated from

---

* The term *nominal horse-power* is no longer used as a unit of measurement, and therefore no mention is made of it here. If the power of a marine engine is given, for brevity sake, as H.P., indicated horse-power is always meant.

† The unit of steam pressure is taken at 1 lb. per square inch.

Middendorf (*Schiffswiderstand und Maschinenleistung*) gives the values of η, or what is known as the *efficiency*:—

TABLE No. 1.

I.H.P.	η = Efficiency.	I.H.P.	η = Efficiency.	I.H.P.	η = Efficiency.
Below 10	0·58	400– 500	0·68	1000–2000	0·79
10– 50	0·59	500– 600	0·69	2000–3000	0·85
50–100	0·60	600– 700	0·71	3000–4000	0·88
100–150	0·61	700– 800	0·72	4000–5000	0·89
150–200	0·62	800– 900	0·73	5000–6000	0·90
200–300	0·64	900–1000	0·74	6000 and over	0·91
300–400	0·66				

To measure the actual work done by means of a brake has hitherto been very difficult in the case of marine engines, owing to the large power to be absorbed, and the difficulties in fitting the brake; until recently the above approximate values were alone available. The latest trials made according to the method described in Part VI. have given the following values:—

In a 1630 I.H.P. engine $\eta = 0{\cdot}885$
„ 1640 „ $\eta = 0{\cdot}91$
„ 1940 „ $\eta = 0{\cdot}911$
„ 2370 „ $\eta = 0{\cdot}92$
„ 2690 „ $\eta = 0{\cdot}911$
„ 4500 „ $\eta = 0{\cdot}935$

§ 2. **Measurement of Indicated Horse-power.**—This is done by means of an instrument called an *indicator*,* with the construction and mode of working of which the reader is supposed to be familiar.

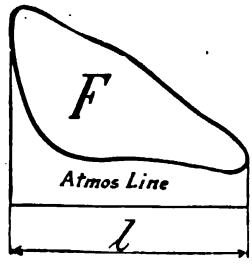


Fig. 1.

As the abscissæ of the indicator diagram are proportional to the volume passed through by the piston, and the ordinates proportional to the steam pressures, the area of the diagram will give the work done during one stroke.

Let F be the area of the indicator diagram in square inches, m = the scale of the indicator spring (so adjusted that if the scale = $\frac{1}{100}$th, the indicator pencil moves

* For the indicator, see Part VI.

one hundredth of an inch for a pressure of 1 lb. per square inch in the steam cylinder), and l the length of the indicator diagram in inches, then the ***mean pressure*** in the cylinder during the stroke, in pounds per square inch, will be—

$$p_m = \frac{F}{l} \div m \text{ (see Fig. 1).}$$

Separate diagrams are taken for the top and the bottom of the cylinder, and the mean of both mean pressures is used to calculate the work done.

The ***atmospheric line*** is the line traced by the indicator pencil when the indicator piston is not connected to the steam cylinder, but is open to the air. Upon each diagram should be marked, whether it was taken from the top or from the bottom of the cylinder, the atmospheric line, the number of the engine, the date and time of the trial, the number of revolutions, the number or allocation of the cylinder (if the engine has more than one), the cut-off, the scale, and lastly the mean pressure and the corresponding indicated horse-power (see Fig. 2).

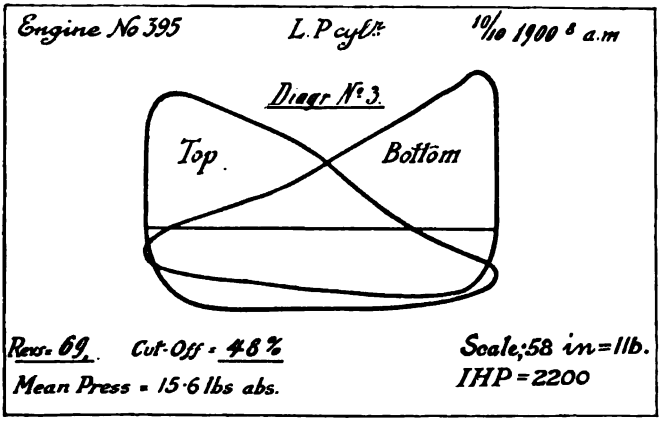


Fig. 2.

The simplest way of determining the mean pressure is by means of a ***planimeter*** (see Part VI.). If, however, none is available, the best way is to use the "Rule of Mean Ordinates," or, if greater accuracy is required, what is known as "Simpson's" formula.

Rule of Mean Ordinates.—Divide the diagram into ten equal parts by lines at right angles to the atmospheric line, and measure the distance in the centre of each division between the top and bottom lines forming the diagram. The mean height of the ten divisions, measured in inches

and divided by the scale of the spring m, is equal to the mean pressure in pounds per square inch (see Fig. 3)—

$$p_m = \frac{h_1 + h_2 + h_3 \ldots h_{10}}{10 \times m}$$

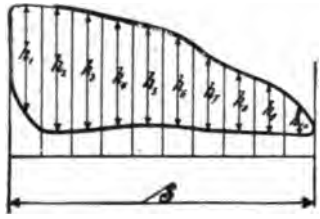


Fig. 3.

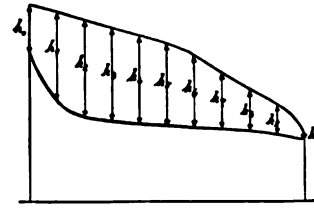


Fig. 4.

Simpson's Formula (see Fig. 4).—Divide the diagram into ten equal parts as before, then if the dividing lines be lettered as shown, and we take—

$$h_0 + h_{10} = H_1,$$
$$h_1 + h_3 + h_5 + h_7 + h_9 = H_2,$$
$$\text{and } h_2 + h_4 + h_6 + h_8 = H_3,$$

the mean pressure (in pounds per square inch) will be—

$$p_m = \frac{H_1 + 4H_2 + 2H_3}{30 \times m}$$

It is important to note that the number of the spaces must be even.

Table of Constants.—The work done in the cylinder in I.H.P. is—

$$\text{I.H.P} = \frac{p_m \times 2s \times A \times n}{33{,}000}, \text{ in which } \frac{2sA}{33{,}000} = C, \text{ a constant.}$$

The constant C is a "characteristic" of each cylinder. If the product $n \cdot C$ be calculated for any number of revolutions, and arranged in the form of a table, the I.H.P. can be quickly obtained. Such tables are frequently employed, especially on trial trips.

In calculating the area of cylinders, the sectional area of the piston rod is sometimes deducted, but in large engines the work is usually calculated without this deduction. If the rod is to be taken into account, and it passes through both cylinder covers, the mean sectional area above and below the piston is usually taken, as it is only the mean power developed in the top and bottom of the cylinder which is used as a basis of calculations.

§ 3. **Measurement of the actual Work exerted by the Engine.**—In larger marine engines it has only been possible quite

recently to do this. Use is made of the fact that all shafting twists, *i.e.*, is distorted under the action of the turning moment exerted by the engine. Two lines which were originally parallel in two different sections of the shaft, in consequence of the torsional pressure, lie at an angle to one another, forming what is known as the torsional angle. So long as the stress upon the material of which the shaft is composed is within the limit of elasticity of the material, this angle is proportional to the turning moment exerted. Let s be the arc of deflection of the angle, measured at a distance R from the centre of the shaft, M the turning moment transmitted to the shaft in inch pounds, L length of the shaft in inches, and I the moment of inertia of the section of the shaft in inches^4, G the modulus of elasticity of the material of the shaft in inch tons, then—

$$s = \frac{M \cdot L \cdot R}{G \cdot I} = \text{constant} \times M.$$

For the material generally used in shafting (Siemens-Martin steel of about 28 tons per square inch tensile strength, and 20 % elongation), G, according to the latest experiments, is = 5,250 inch tons. As the mean turning moment $M = \frac{198{,}000}{\pi} \cdot \frac{N}{n}$, where N = work done in H.P., n = revolutions of the engine per minute, we get—

$$s = \frac{198{,}000}{\pi} \cdot \frac{N}{n} \cdot \frac{L \cdot R}{5{,}250 \times 2{,}240\, I} = \frac{198{,}000}{5{,}250 \times 2{,}240\pi} \cdot \frac{N}{n} \cdot \frac{L \cdot R}{\frac{d^4}{32}}$$

$$= \cdot 0546 \cdot \left(\frac{N}{n}\right) \cdot \frac{L \cdot R}{d^4}$$

= mean arc of deflection during one revolution.

In *turbines* and *electric motors*, the turning moment M, and hence the arc of deflection s, is a constant, and equal to the corresponding mean value.

In *reciprocating engines* the turning moment passes, at each revolution, through a range of fluctuations; as the arc of deflection s = constant × M, it varies in proportion to the turning moment. If the arc of deflection be measured at different positions of the crank, the corresponding actual turning moment can easily be determined from it; the curve of the arcs of deflection plotted above the developed circle of the crank represents, on a different scale, the curve of actual turning moments, or tangential pressures.

The arcs of deflection may be *determined experimentally* in either of the following ways:—

1. In long shafts (from 60 to 100 feet in length), by utilising the

instantaneous action of the electric current (by Frahm's,* Professor Denton's, and Föttinger's methods †).

2. By the very latest method in which a torsional indicator records automatically the torsional deflections of the shaft, and draws a diagram giving the actual tangential pressures.

For further details, see Measuring Apparatus, Part VI.,
also Shafting, Part III.,
and Torsional Vibrations, page 74.

Let s be the mean arc of deflection obtained from the diagrams, and n the mean number of revolutions of the engine, then from the foregoing equation we get—

$$\text{Mean turning moment } M = s \cdot \frac{G \cdot I}{L \cdot R} \text{ inch pounds,}$$

and the actual work of the engine—

$$\text{B.H.P.} = \frac{M \cdot n}{63{,}020} = s \cdot n \cdot \frac{d^4}{\cdot 0546 \cdot L \cdot R} = s \cdot n \cdot \text{constant.}$$

To obtain the B.H.P. quickly it is advisable to draw up a table giving the values of $n \times$ constant for the various speeds at which the engine is likely to run.

§ 4. **Indicator Diagrams and Steam Distribution** (see Fig. 5).—The steam enters the cylinder at a pressure somewhat less than that of the boiler p, and fresh steam is admitted while the piston travels along part of its stroke a, which is called the *period of admission;* the steam then expands along the part of the stroke e, which is called the *period of expansion;* before the end of the stroke, while the piston is passing from e to g, the *exhaust* opens, and the steam is discharged while the piston travels back along part of the stroke h—this is the *period of exhaust;* the exhaust closes, and the steam remaining in the cylinder is compressed along the part of the stroke c, which is the *period of compression;* a little before the piston reaches the dead point, at v, fresh steam is admitted—*admission.*‡

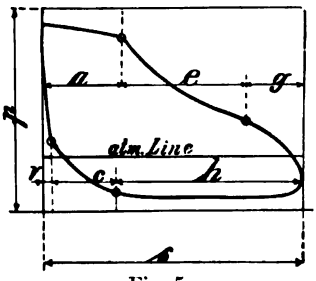


Fig. 5.

For description of the manner in which this distribution of the

* *Zeitschrift des Vereines Deutscher Ingenieure*, 1902.

† *Jahrbuch des Schiffbautechnischen Gesellschaft*, 1903.

‡ In future the letters a, e, g, s, &c., in the diagram Fig. 5, will be used to denote the corresponding cylinder volumes. Thus s signifies the volume of the entire stroke in the cylinder, a volume at cut-off, &c.

steam is obtained, and of the way in which it should be carried out, see "Valves," Section 72.

§ 5. **Compound Expansion.**—*Single-cylinder engines* are those in which the whole work of the steam is performed in one cylinder.

Twin-cylinder engines are those in which each cylinder works in the same way as a single-cylinder engine; the steam passes into both cylinders direct from the boiler, and out of both direct to the condenser, or to the atmosphere.

Compound engines are those in which the steam works successively in several cylinders placed close to each other. It passes from the boiler into the high-pressure (H.P.) cylinder, from thence—

(*a.*) In a *two-cylinder compound* engine to the low-pressure (L.P.) cylinder, and so on to the condenser, or to the atmosphere.

(*b.*) In *triple expansion engines* to the intermediate cylinder, thence to the low-pressure cylinder, and then to the condenser, or atmosphere.

(*c.*) In *quadruple expansion engines* to an intermediate cylinder, then to a second larger intermediate cylinder, and thence to the low-pressure cylinder, and so on to the condenser.

As the steam decreases in pressure in its passage through the various cylinders, and increases correspondingly in volume (see Table XXII., Part VIII.), the sizes of the cylinders, from the high-pressure cylinder onwards, must be larger according to the degree of expansion employed.

For constructive reasons the same degree of expansion may, in large engines, be sometimes divided between two cylinders—either high or low pressure—which are placed side by side, as in "twin-cylinder" engines. Thus it often happens that in a triple expansion engine there are five cylinders, namely, two H.P., one intermediate, and two L.P. (see "Arrangements of Main Engines," page 106). The cylinder dimensions are calculated as if there were only *one* cylinder for each degree of expansion, equal in volume to the combined volumes of the two cylinders.*

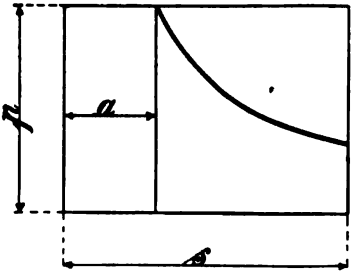


Fig. 6.

Compound engines are calculated in precisely the same way as single-cylinder engines; the reasoning is the same as if all the work of the steam were done in the low-pressure cylinder.

§ 6. **Work of the Steam in the Cylinder.**

(1.) *Admission.*—Assuming that there is no clearance, and that

* For the reason why compound expansion is employed, see page 37.

during the period of admission the pressure in the cylinder is the same as that in the boiler p, the work done by the steam during admission (see Fig. 6) is—

$$\text{Work of admission} = a \,.\, p.$$

The mean pressure during this period, in relation to the whole stroke, is thus—

$$p_a = p \,.\, \frac{a}{s}$$

The quotient $\frac{a}{s} = \epsilon$ is called the *cut-off*, which is represented either as a fraction, or as a percentage of the volume of the cylinder s. The reciprocal value $\frac{s}{a} = \frac{1}{\epsilon}$ is called the *degree of expansion*.

In compound engines, the term *total cut-off* is understood to mean the ratio that the volume of steam admitted to the high-pressure cylinder bears to the volume of the low-pressure cylinder; by the term *total expansion* is meant the reciprocal of this ratio.

If m be the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder, and ϵ_h denote the cut-off in the H.P. cylinder, we get—

$$\text{Total cut-off} \qquad \epsilon = \frac{a}{m \,.\, s} = \frac{\epsilon_h}{m}$$

$$\text{Total expansion} \qquad \frac{1}{\epsilon} = \frac{m \,.\, s}{a} = m \,.\, \frac{1}{\epsilon_h}$$

For remarks on the selection of the proper proportions of the cylinders, see page 21.

For information respecting the proper total degree of expansion, see page 22 *et seq.*

(2.) *Expansion.*—After the valve has cut off the flow of fresh steam to the cylinder, the steam in the cylinder begins to expand, and drives the piston before it.

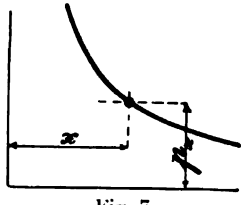


Fig. 7.

A study of actual diagrams shows that the expansion line usually resembles very closely a hyperbola,* and hence this curve is usually taken as the basis for calculating the work done during expansion, the more so as it is easily calculated. The equation (see Fig. 7) is—

$$p_x \,.\, x = \text{constant},$$

i.e., the product of the volume and the pressure of the expanding steam at each point of the stroke is a constant.†

* For the actual nature of the expansion curve, see page 33.

† Care must be taken not to confuse this purely theoretical curve of expansion with

Construction of the rectangular hyperbola used as the curve of expansion (Fig. 8). Let AB = p, the initial pressure; BC = a, the volume up to cut-off; AG = s, the volume at the end of the stroke.

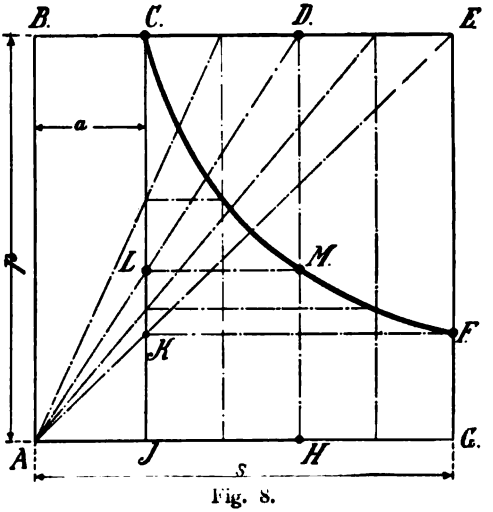


Fig. 8.

To get the final pressure after expansion, draw the diagonal line AE; then through the point K, the intersection of CJ and AE, draw the line KF parallel to AG. The line FG gives the required final pressure. To get the pressure at any given point of the expansion curve, say for the volume BD = AH, draw the diagonal line AD, then LM parallel to AG; the point M where it intersects the line DH is the required point in the expansion curve, the line MH giving the pressure corresponding to the volume AH.

To ascertain what volume the quantity of steam a at the pressure p would occupy if it were compressed to the pressure p_1, draw the diagonal line AP (Fig. 9), then QO parallel to NA. The line NO gives the volume required.

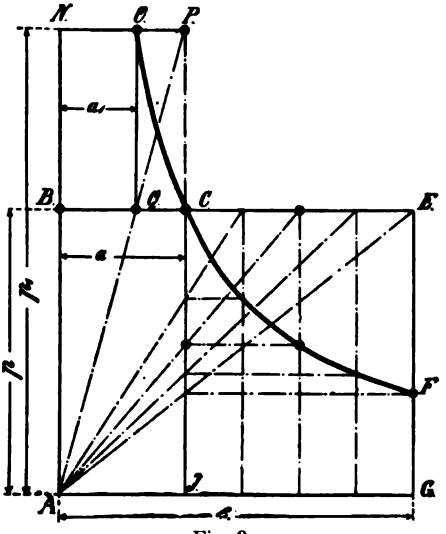


Fig. 9.

Work of Expansion.—This is equal to the area CFGJ, below the expansion curve (Fig. 10).

We have CFGJ $= \int_a^s p_x dx$.

Since the product of the

---

the isothermal expansion curve of a perfect gas, which is also a rectangular hyperbola. Expansion in a steam-engine cylinder is not isothermal, as the temperature of the steam falls as it expands.

volumes and pressures is a constant for each point of the expansion curve—

$$p_x \cdot x = p \cdot a, \text{ and } p_x = \frac{p \cdot a}{x}$$

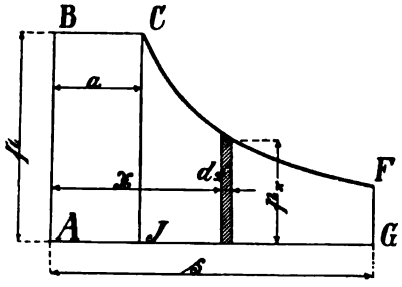


Fig. 10.

Therefore CFGJ $= \int_a^s \frac{p \cdot a}{x} \cdot dx = pa \left[\log_e x\right]_a^s = p \cdot a \log_e \frac{s}{a}$

The mean pressure during the work of expansion in relation to the stroke s is therefore—

$$p_e = p \cdot \frac{a}{s} \log_e \frac{s}{a}$$

$$= p \cdot \epsilon \log_e \frac{1}{\epsilon}$$

(3.) *Mean Theoretical Pressure of Admission and Expansion.*—This value is frequently used as a basis of calculation when determining the dimensions of cylinders. It is taken from what is called a *theoretical diagram* (see Fig. 10). The admission pressure is assumed to be the boiler pressure in pounds per square inch absolute. The hyperbola is used for the expansion curve, and perfect vacuum for the exhaust pressure. The mean pressure of such a diagram is—

$$p_t = p_a + p_e = p \cdot \epsilon + p \cdot \epsilon \log_e \frac{1}{\epsilon} = p \cdot \epsilon \left[1 + \log_e \frac{1}{\epsilon}\right]$$

If the expansion takes place in a compound engine, ϵ signifies the total cut-off, $\frac{1}{\epsilon}$ the total expansion.

Table No. 3 gives the ratio $\frac{p_t}{p}$ for different values of ϵ and $\frac{1}{\epsilon}$

TABLE No. 3.

Mean Theoretical Pressures of Steam.

$\frac{1}{\epsilon}$	ϵ	$\frac{p_t}{p}$	$\frac{1}{\epsilon}$	ϵ	$\frac{p_t}{p}$	$\frac{1}{\epsilon}$	ϵ	$\frac{p_t}{p}$
1·33	0·752	0·9657	4·6	0·217	0·5491	7·8	0·128	0·3915
1·4	0·714	0·9546	4·7	0·213	0·542	7·9	0·127	0·3882
1·5	0·667	0·937	4·75	0·211	0·5385	8·0	0·125	0·3849
1·6	0·625	0·9188	4·8	0·208	0·5351	8·1	0·123	0·3817
1·7	0·588	0·9003	4·9	0·204	0·5284	8·2	0·122	0·3785
1·75	0·571	0·8911	5·0	0·200	0·5219	8·25	0·121	0·377
1·8	0·556	0·882	5·1	0·196	0·5155	8·3	0·120	0·3755
1·9	0·526	0·8641	5·2	0·192	0·5093	8·4	0·119	0·3724
2·0	0·500	0·8465	5·25	0·190	0·5063	8·5	0·118	0·3694
2·1	0·476	0·8294	5·3	0·189	0·5033	8·6	0·116	0·3665
2·2	0·455	0·8129	5·4	0·185	0·4975	8·7	0·115	0·3636
2·25	0·444	0·8048	5·5	0·182	0·4917	8·75	0·114	0·3622
2·3	0·435	0·7968	5·6	0·179	0·4862	8·8	0·114	0·3608
2·4	0·417	0·7814	5·7	0·175	0·4808	8·9	0·112	0·358
2·5	0·400	0·7665	5·75	0·174	0·4781	9·0	0·111	0·3552
2·6	0·385	0·7521	5·8	0·172	0·4755	9·1	0·110	0·3526
2·7	0·370	0·7382	5·9	0·169	0·4703	9·2	0·109	0·3499
2·75	0·364	0·7315	6·0	0·167	0·4652	9·25	0·108	0·3486
2·8	0·357	0·7249	6·1	0·164	0·4604	9·3	0·108	0·3473
2·9	0·345	0·712	6·2	0·161	0·4555	9·4	0·106	0·3447
3·0	0·333	0·6995	6·25	0·160	0·4532	9·5	0·105	0·3422
3·1	0·323	0·6875	6·3	0·159	0·4509	9·6	0·104	0·3396
3·2	0·313	0·676	6·4	0·156	0·4463	9·7	0·103	0·3373
3·25	0·308	0·6703	6·5	0·154	0·4418	9·75	0·103	0·3361
3·3	0·303	0·6648	6·6	0·152	0·4374	9·8	0·102	0·3349
3·4	0·294	0·654	6·7	0·149	0·4331	9·9	0·101	0·3326
3·5	0·286	0·6436	6·75	0·148	0·431	10·0	0·100	0·3302
3·6	0·278	0·6335	6·8	0·147	0·4289	10·1	0·099	0·3279
3·7	0·270	0·6238	6·9	0·145	0·4248	10·2	0·098	0·3257
3·75	0·267	0·6191	7·0	0·143	0·4208	10·25	0·097	0·3246
3·8	0·263	0·6145	7·1	0·141	0·4169	10·3	0·097	0·3224
3·9	0·256	0·6054	7·2	0·139	0·4131	10·4	0·096	0·3213
4·0	0·250	0·5965	7·25	0·138	0·4111	10·5	0·095	0·3191
4·1	0·244	0·588	7·3	0·137	0·4093	10·6	0·094	0·3170
4·2	0·238	0·5798	7·4	0·135	0·4056	10·7	0·093	0·315
4·25	0·235	0·5757	7·5	0·133	0·4019	10·75	0·093	0·314
4·3	0·233	0·5717	7·6	0·132	0·3984	10·8	0·093	0·3129
4·4	0·227	0·564	7·7	0·130	0·3949	10·9	0·092	0·3109
4·5	0·222	0·5564	7·75	0·129	0·3932	11·0	0·091	0·3088

TABLE No. 3—*continued.*

$\frac{1}{\epsilon}$	ϵ	$\frac{p_1}{p}$	$\frac{1}{\epsilon}$	ϵ	$\frac{p_1}{p}$	$\frac{1}{\epsilon}$	ϵ	$\frac{p_1}{p}$
11·1	0·090	0·3065	12·75	0·0784	0·2781	14·5	0·0690	0·2534
11·2	0·089	0·3049	12·8	0·0781	0·2773	14·6	0·0685	0·2521
11·25	0·089	0·304	12·9	0·0775	0·2757	14·7	0·0680	0·2509
11·3	0·0885	0·3030	13·0	0·0769	0·2741	14·8	0·0676	0·2496
11·4	0·0877	0·3011	13·1	0·0763	0·2726	14·9	0·0671	0·2484
11·5	0·0870	0·2994	13·2	0·0758	0·2712	15·0	0·0667	0·2472
11·6	0·0862	0·2974	13·25	0·0755	0·2705	15·1	0·0662	0·2460
11·7	0·0855	0·2956	13·3	0·0752	0·2697	15·2	0·0658	0·2448
11·75	0·0851	0·2947	13·4	0·0746	0·2683	15·3	0·0654	0·2436
11·8	0·0847	0·2939	13·5	0·0741	0·2668	15·4	0·0649	0·2425
11·9	0·0840	0·2921	13·6	0·0735	0·2654	15·5	0·0645	0·2413
12·0	0·0833	0·2904	13·7	0·0730	0·2640	15·6	0·0641	0·2402
12·1	0·0826	0·2887	13·75	0·0727	0·2633	15·7	0·0637	0·2391
12·2	0·0820	0·287	13·8	0·0725	0·2626	15·8	0·0633	0·238
12·25	0·0816	0·2861	13·9	0·0719	0·2613	15·9	0·0629	0·2369
12·3	0·0813	0·2853	14·0	0·0714	0·2599	16·0	0·0625	0·2358
12·4	0·0806	0·2836	14·1	0·0709	0·2586	16·25	0·0615	0·2331
12·5	0·0800	0·2821	14·2	0·0704	0·2573	16·50	0·0606	0·2305
12·6	0·0794	0·2804	14·3	0·0699	0·256	17·0	0·0588	0·2255
12·7	0·0787	0·2789	14·4	0·0694	0·2547			

For instructions as to the use of this table, see page 19.

For fuller information on *exhaust*, *compression*, *lead*, and *release*, see "Valves."

§ 7. **Clearance.**—Before the incoming steam can force the piston out, it has to fill the space between the piston and the valve face, known as the "clearance" space. This space has an injurious effect on the working and economy of the engine, because it is filled alternately with hot admission steam and with exhaust steam of much lower temperature, and therefore the large superficial area of ports between the piston and cover causes considerable losses by condensation during admission. For further disadvantages, see page 36.

It prejudicially affects expansion, because it raises the terminal pressure, and also affects compression, because it reduces the final pressure of compression.

In Fig. 11 the expansion curves are drawn with and without taking the clearance σ into account; if it be included the point A′ forms the starting point for the expansion line, because the steam in the clearance is also expanded.

The work of expansion is increased by the clearance—

$$JCF'G > JCFG.$$

(JCFG being work of expansion without the clearance.)

It can thus be proved that the compression, necessary to obtain a given terminal pressure, must be increased, if there is a clearance space.

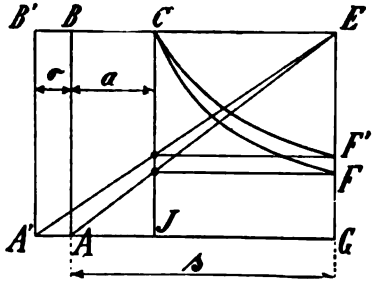


Fig. 11.

The clearance σ is measured in fractions, or as a percentage, of the whole cylinder volume of which it forms a part.

The *mean pressure*, including admission and expansion, and taking account of the clearance, is, according to the reasoning on page 12—

$$p_\sigma = p \cdot \frac{1}{s}\left[a + (a+\sigma)\log_e\left(\frac{s+\sigma}{a+\sigma}\right)\right]$$

How far the clearance affects the mean "theoretical" pressure, is shown in the next table (No. 4). Clearance is always considerable in marine engines. In cylinders with slide-valves it is—

For large cylinders, from 8 to 14 °/₀.
For small cylinders, from 10 to 15 °/₀.

The higher values may be used for quick-running engines, with large ports. In large low-pressure cylinders of mercantile steamers, fitted with flat slide-valves, the clearance is generally from 8 to 10 °/₀ of the cylinder volume.

In cylinders with piston valves the clearance may be taken as—

12 to 18 °/₀ for small cylinders with short straight ports.
15 to 19 °/₀ for very large cylinders with long ports.
18 to 30 °/₀ for small and medium sized cylinders (H.P. and intermediate cylinders of war-vessels), with long and large ports. The higher values may also be used for quick-running engines.

TABLE No. 4.

Values of $\frac{p_\sigma}{p}$ (*Mean Theoretical Pressure, taking Clearance into Account*).

(From Haeder, *Die Dampfmaschinen.*)

Admission or Cut-off ϵ	Percentage of Clearance Volume.						
	0 %	6 %	8 %	10 %	12 %	14 %	16 %
0·00	0·00	0·17	0·21	0·24	0·27	0·29	0·32
0·05	0·19	0·30	0·32	0·35	0·37	0·39	0·41
0·06	0·22	0·32	0·34	0·37	0·39	0·41	0·42
0·07	0·25	0·34	0·36	0·39	0·41	0·43	0·44
0·08	0·28	0·36	0·38	0·40	0·42	0·44	0·46
0·09	0·31	0·38	0·40	0·42	0·44	0·46	0·48
0·10	0·33	0·40	0·42	0·44	0·46	0·47	0·49
0·12	0·37	0·44	0·45	0·47	0·49	0·50	0·52
0·14	0·42	0·47	0·49	0·50	0·52	0·53	0·54
0·16	0·46	0·50	0·52	0·53	0·55	0·56	0·57
0·18	0·49	0·53	0·55	0·56	0·57	0·59	0·60
0·20	0·52	0·57	0·58	0·59	0·60	0·61	0·62
0·25	0·60	0·63	0·64	0·65	0·66	0·67	0·68
0·30	0·67	0·69	0·70	0·70	0·71	0·72	0·73
0·40	0·77	0·78	0·79	0·80	0·80	0·80	0·81
0·50	0·84	0·86	0·86	0·87	0·87	0·87	0·88
0·60	0·90	0·91	0·91	0·92	0·92	0·92	0·92

§ 8. **Calculation of Cylinder Dimensions for a given Horse-power.**—The power developed by an engine in indicated horse-power is (see page 3)—

$$\text{I.H.P.} = \frac{p_m \times 2s \times A \times n}{33{,}000}$$

$$\text{Where } A = \frac{\pi D^2}{4}$$

If the stroke, number of revolutions, and cut-off be assumed, the diameter of the cylinder can be calculated as soon as the actual mean pressure p_m has been determined. The calculation of the cylinder dimensions thus depends wholly on the determination of p_m.

In compound engines the whole work of the steam is assumed to be carried out in the L.P. cylinder. The mean pressure is determined for a

corresponding single-cylinder engine, and is inserted in the formula for the I.H.P., in which D denotes the diameter of the L.P. cylinder.

§ 9. **Determination of the Actual Mean Pressure (p_m) for a New Engine.**—This may be done by determining the mean pressure p_t from the theoretical or empirical diagram, and reducing it to the value p_m by comparing it with results obtained from similar engines.

§ 10. **The Theoretical Diagram and Efficiency** (k).—This diagram shows how much work the steam admitted at each stroke from the valve chest of the H.P. cylinder could do, if it were expanded from its initial to its final volume. Here it is assumed—

1. That the whole clearance is filled at each stroke.
2. That the expansion curve is hyperbolic.
3. That the steam exhausts from the L.P. cylinder into a perfect vacuum.
4. That there is no compression, no lead, and no premature opening to exhaust.
5. That no losses by condensation, leakage, &c., take place.

The steam pressure at the point of cut-off in the H.P. cylinder is always lower than in the boiler, but in this diagram during the whole period of admission it should be taken at full boiler pressure. The construction of the diagram is based on the assumption that the admission is at the pressure p. The diagram is obtained by lengthening the hyperbola C′F′ along the line to C (Fig. 12). The distance BC = a represents the desired reduced period of cut-off. The volume $a_1 + \sigma$ is the initial volume before expansion. The quantity of steam $a_1 + \sigma$ at the pressure p is the same as the quantity $a + \sigma$ at the initial pressure of admission in the H.P. cylinder. Fig. 12 shows the theoretical diagram of a single-cylinder engine, with the actual diagram of the engine sketched-in inside. On the above assumptions the area A′B′CF′G of the theoretical diagram represents the energy contained in the quantity of steam $\sigma + a_1$.

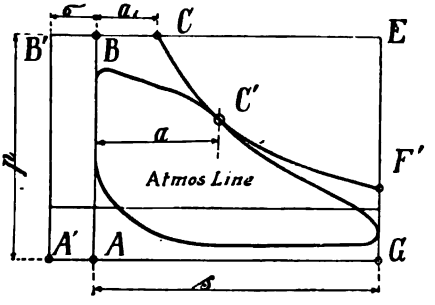


Fig. 12.

The theoretical diagram of a compound engine is determined in the same way. The initial volume, as already stated, is equal to the

clearance in the H.P. cylinder + its volume up to point of cut-off; the final volume is equal to the clearance in the L.P. cylinder + the total volume of the L.P. cylinder.

Fig. 13 (Plate I.) shows the *theoretical combined diagram* of a triple engine. The area A′B′CF′G is the total area of the diagram; the distance B′C is the volume of the steam at cut-off, reduced to the pressure of the boiler (during the period of admission), + clearance in the H.P. cylinder. The diagram of the actual engine is sketched in inside the other. The figure obtained by drawing the actual indicator diagrams inside the theoretical diagram is called a *combined or Rankine diagram.*

§ 11. **Combining the Diagrams.**—When the two are thus shown together, the clearance belonging to each cylinder is set off from A′B′; and the expansion curves of the separate diagrams will then lie correctly as they are affected by the clearance (see above). The lengths of the diagrams are to each other as the volumes of their respective cylinders. The ratio of the areas of the actual diagram $F_1 + F_2 + F_3$ to the area of the theoretical diagram A′B′CF′G is called the **efficiency** k.

If p_m denotes the sum of the mean pressures in the different cylinders, reduced to the L.P. cylinder, then—

$$\text{Efficiency } k = \frac{F_1 + F_2 + F_3}{A'B'CF'G} = \frac{p_m \cdot s}{p_t \cdot s} = \frac{p_m}{p_t}$$

If, when designing an engine, the mean pressure p_t be determined with a planimeter from the theoretical diagram, it is only necessary to multiply it by the efficiency k, to get the requisite mean pressure p_m of the actual engine. Approximate values for k may be obtained, as already stated, by drawing the actual indicator diagrams inside the theoretical diagram. When choosing an efficiency, that of a known engine of the same type, and, as far as possible, of the same dimensions, should be selected.

To construct a theoretical diagram for a new engine, it is necessary to know the fall in pressure of the steam before expansion begins. This depends on the size and length of the steam pipes, their position, loss by radiation, the number of revolutions, arrangement of the passages in the H.P. cylinder, and the type of valve gear.

The following table gives the data for determining this fall in pressure:—

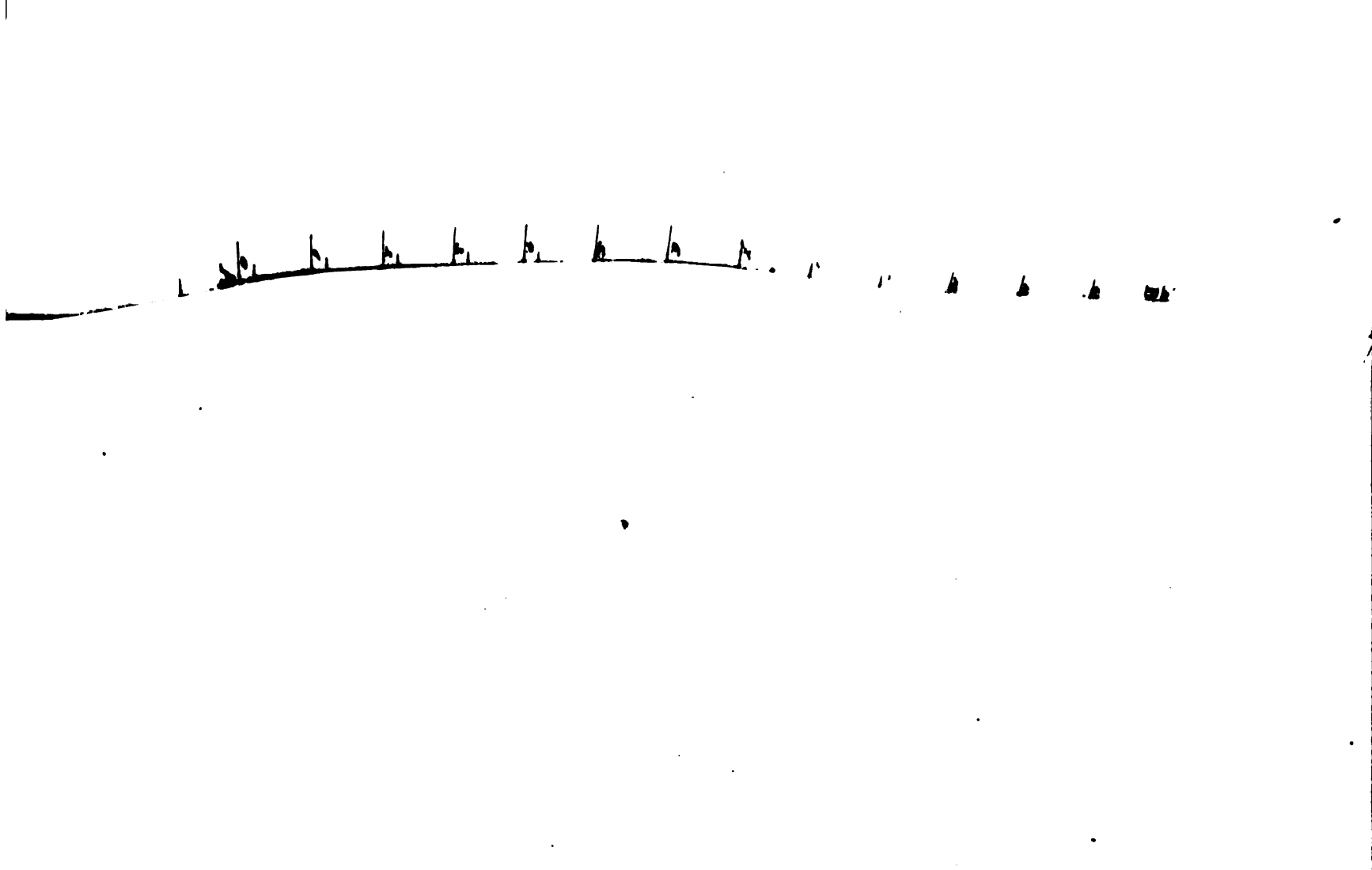


TABLE No. 5.

Fall in Pressure of the Steam between the Boiler and the beginning of Expansion (between C *and* C').

Small single-cylinder and twin-cylinder engines - p = 70 to 100 lb. per square inch - - - - -	12 to 17 lb. per square inch for low boiler pressures and low speeds. 17 to 22 lb. per square inch for high boiler pressures and high speeds.
Compound engines - - p = 100 to 150 lb. per square inch - - - - -	22 to 30 lb. per square inch for low boiler pressures and low speeds. 30 to 45 lb. per square inch for high boiler pressures and high speeds.
Large triple and quadruple expansion engines - - p = at or above 160 lb. per square inch - - -	50 to 60 lb. per square inch.
Large triple and quadruple expansion engines - - p = at or above 220 lb. per square inch - - -	52 to 65 lb. per square inch.

§ 12. **In Designing Engines and for Approximate Calculations** the mean theoretical pressure p_t is not taken from the combined diagram, but simply from Table No. 3 on page 13. To get the actual mean pressure, p_t is multiplied by the efficiency k. Fig. 14 shows how illogical this process is. The efficiency is, as we have said—

$$k = \frac{F_1}{A'B'CF'G}$$

If the method of calculation given above be used, it works out as though it were—

$$k = \frac{F_1}{ABDFG}$$

The reason why this method gives fairly satisfactory results is that the shaded areas A'B'BA and CDFF' are, under ordinary conditions, of practically the same size.

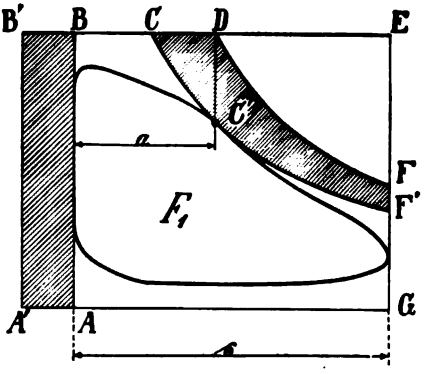


Fig. 14.

The *mean theoretical pressure* will in future be denoted by (p_t),

whether obtained by the more exact method, or only by the approximate one.*

For an example of method of designing a Triple Engine, by means of the theoretical diagram, see page 24.

TABLE No. 6.

Efficiency k, for Condensing Engines.

The values for k given below hold for the usual number of total expansions and cylinder ratios.

Expansion in a Single Cylinder.	
Large slow-speed engines	$k = 0{\cdot}70$ to $0{\cdot}75$
Small high-speed engines	$k = 0{\cdot}65$ „ $0{\cdot}70$
Expansion in a Two-Cylinder or Compound Engine.	
Large engines up to about 100 revolutions per minute	$k = 0{\cdot}60$ „ $0{\cdot}67$
Small engines, with a higher number of revolutions	$k = 0{\cdot}55$ „ $0{\cdot}60$
Triple Expansion in Three Cylinders.	
War-vessels, with a high number of revolutions	$k = 0{\cdot}53$ „ $0{\cdot}54$
Mercantile ships, up to 100 revolutions per minute	$k = 0{\cdot}56$ „ $0{\cdot}61$†
Triple Expansion in Four or Five Cylinders.	
High-speed engines	$k = 0{\cdot}50$ „ $0{\cdot}52$
Mercantile steamers, up to 100 revolutions per minute	$k = 0{\cdot}54$
Quadruple Expansion in Four or more Cylinders.	
Large mercantile steamers	$k = 0{\cdot}52$ „ $0{\cdot}53$

N.B.—The mean of the indicator diagrams, taken from the top and bottom of the cylinders, has been used to draw up the above table.

For non-condensing engines the mean pressure, calculated from the above data, must be reduced by about 15 lb. per square inch, on account of the back pressure.

---

* The method here described for constructing the theoretical diagram is only to be used for obtaining comparative results. It is inaccurate to assume that the clearance of each cylinder is filled afresh at every stroke. This inaccuracy has purposely been allowed to stand, to avoid further complications of the diagram, and because much greater inaccuracies would inevitably arise in selecting the coefficient k for new calculations. The method has also been retained, because it is in general use among marine engineers, and gives fairly good results.

† Rarely over 0·58.

§ 13. **Number of Expansions, Cylinder Ratios, and Cut-off in each Cylinder.**—Expansion can only be carried to such a point that the final or terminal pressure in the L.P. cylinder is equal to the exhaust pressure. This lower limit can easily be determined from the theoretical diagram, but it is never realised in actual practice. Expansion is limited in this direction by the question of economy, which falls off if the expansion is carried too far. In engines for warships and small light craft, the maximum power is based upon a comparatively small number of expansions, the reason being that in the former type of vessel the highest power has only to be developed occasionally.

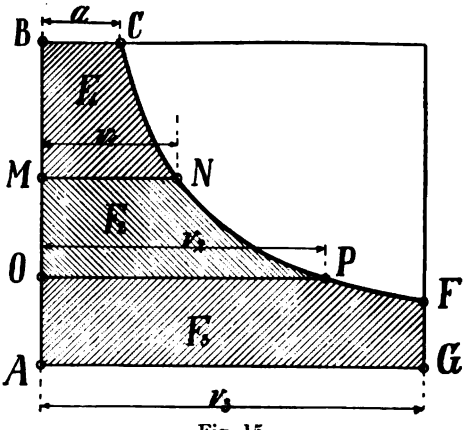


Fig. 15.

In § 14, page 22, the normal number of expansions for different types of engines are given. The theoretical diagram, Fig. 15, gives a starting point for determining the *ratio of the cylinder dimensions* and the *cut-off*, in the intermediate and L.P. cylinders.

If the area of the diagram ABCFG represents the actual work done in a compound engine, the cylinder dimensions and cut-off corresponding to the work in each cylinder would be determined thus :—

v_3 = volume of the L.P. cylinder; and

$\frac{a}{v_3}$ = total admission or cut-off being assumed :—

ABCFG is divided into three equal areas, F_1 F_2 and F_3 by drawing parallel lines to AG.

Then v_1 = volume of the H.P. cylinder.

$\frac{a}{v_1}$ = cut-off or admission in the H.P. cylinder.

v_2 = volume of the intermediate cylinder.

$\frac{v_1}{v_2}$ = cut-off or admission in the intermediate cylinder.

$\frac{v_2}{v_3}$ = cut-off or admission in the L.P. cylinder.

$1 : \frac{v_2}{v_1} : \frac{v_3}{v_1}$ = ratio of cylinders.

The areas F_1, F_2, F_3 do not in any way represent the actual diagrams.

If account be taken of the clearance (page 14), the back pressure in the condenser, compression, early admission, and other similar data, the fall in pressure of the steam on entering the H.P. cylinder, and also between the various cylinders, the diagrams F_1, F_2, F_3 may be so modified as to resemble very closely the actual diagrams. From their shape or form the dimensions of the cylinders may be determined. The simplest way is to take the cylinder ratio and the cut-off from engines which have given satisfactory results.

In marine engines an endeavour is nearly always made to distribute the power equally on each crank, but for constructive reasons it is not always possible to do this, or to obtain a uniform turning moment (see page 63) or reciprocation of the moving parts (page 82).*

The cut-off in the separate cylinders may be varied within such wide limits, by adjusting the valve-gear while the engine is running, that the division of the work between the cylinders may be considerably altered. If, for instance, in a triple expansion engine the cut-off in the intermediate cylinder be increased, the exhaust pressure in the H.P. cylinder falls, and more work is thrown upon the H.P. cylinder. As the total work in the three cylinders does not vary considerably so long as the cut-off in the H.P. cylinder remains the same, the work done in the intermediate cylinder will be less as cut-off is increased. In the same way, by making the cut-off in the L.P. cylinder less, more work will be done in that cylinder, and less in the intermediate, where the back pressure will be increased.

An ordinary triple expansion three-cylinder engine may be cited as an example where the work was—

With a cut-off of	70 °/₀	in the	H.P.	cylinder	690 I.H.P.	}	2,170 I.H.P.
,,	,, 71	,,	M.P.	,,	700 I.H.P.		
,,	,, 55	,,	L.P.	,,	780 I.H.P.		
and with a cut-off of	70 °/₀	in the	H.P.	cylinder	690 I.H.P.	}	2,138 I.H.P.
,,	,, 71	,,	M.P.	,,	730 I.H.P.		
,,	,, 60	,,	L.P.	,,	718 I.H.P.		

§ 14. The following may be taken as the **total cut-off** and **cylinder dimensions** for various types of engines :—

* In practice it is hardly ever possible to obtain the same fall or range of temperature in the different cylinders.

1. *Single-cylinder Engines.*—These are hardly ever used now except as twin-cylinder engines for small light boats, steam pinnaces, &c.

Cut-off $\epsilon = 60$ to 80 °/₀.

Auxiliary engines, such as steam cranes, reversing and turning engines, circulating engines, &c., generally work with a very late cut-off, and sometimes even with full admission. This is always the case if the reversing engine is worked by rotary or reciprocating slide valves (see Reversing Engines).

2. *Compound Engines.*—These are now only used for small freight and passenger boats developing up to 300 H.P. The total rate of expansion depends upon the space available and the weight of the engine.

Light Engines.—$p = 100$ to 140 lb. per square inch.*

$$\frac{\text{H.P.}}{\text{L.P.}} = m = 3{\cdot}2 \text{ to } 3{\cdot}8.$$

Heavy Engines.—$p = 90$ to 100 lb. per square inch.

$$m = 4 \text{ to } 4{\cdot}6.$$

Cut-off in H.P. cylinder 50 to 70 °/₀, which corresponds to a total expansion of about $\frac{1}{\epsilon} = 5$ to 8.

For the table giving data from actual compound engines see page 42.

3. *Triple Expansion Engines.*—These are made in many types and sizes. For various arrangements of cylinders see page 106. The cylinder dimensions and rate of expansion vary correspondingly within a wide range.

(a.) *Engines for Torpedo-boats and Torpedo-boat Destroyers.*—$p = 160$ to 220 lb. per square inch, and occasionally up to 250 lb. per square inch. Ratio of the cylinders about 1 : 2·1 : 4·4 up to 1 : 2·2 : 5. Cut-off in H.P. cylinder about 65 °/₀ at maximum power. Total expansion 6·8 to 7·7 times the original volume.

(b.) *Engines for Cruisers, &c.*—$p = 160$ to 220 lb. per square inch

* Here and in future the pressure above the atmosphere is given at the stop-valve of the engine and called (p). In water-tube boilers steam is often generated at from 250 to 300 lb. per square inch, and the pressure is reduced before entering the engine, by means of a reducing valve, to a pressure of from 180 to 225 lb. per square inch.

(above atmosphere), sometimes up to 250 lb. per square inch. Ratio of the cylinders about 1 : 2·3 : 5·5. Cut-off in H.P. cylinder about 70 °/₀ at maximum power. Total expansion about 7·5 to 8 times the original volume.

(c.) *Engines for Ironclads, Fast Steamers, and Mail-boats.*—$p = 160$ to 200 lb. per square inch. Ratio of cylinders about 1 : 2·4 : 6 up to 1 : 3 : 7. Cut-off in H.P. cylinder 70 to 75 °/₀ at maximum power. Total expansion about 8 to 10 times the original volume.

(d.) *Engines for Large Slow Freight and Passenger Steamers, and for Cargo-boats.*—$p = 150$ to 220 lb. per square inch. Ratio of cylinders about 1 : 2·6 : 6·8 up to 1 : 3·2 : 7·2. Cut-off in H.P. cylinder 60 to 70 °/₀ at maximum power. Total expansion about 9·5 to 12 times the original volume.

For table giving data from actual triple expansion engines see page 44 *et seq.*

4. *Quadruple Expansion Engines.*—These are only used where a high rate of expansion can be obtained. On small light ships this is not possible, as weight and space must be economised. Quadruple expansion engines are therefore chiefly found in *fast mail steamers* and *large freight and passenger vessels.* $p = 190$ to 220 lb. per square inch (above atmosphere). Ratio of cylinders 1 : 2 : 4 : 8 up to 1 : 2.2 : 4·4 : 9·2. Cut-off in H.P. cylinder 65 to 72 °/₀ at maximum power. Total expansion about 10 to 13 times the original volume.

For table giving data from actual quadruple expansion engines see page 56.

For arrangement of cylinders of such engines see page 120.

§ 15. **Example of Method of Designing Triple Expansion Engine for a Screw Mail Steamer.**—*Assuming:* I.H.P. = 6,300; $n = 75$ revolutions per minute; $c = 786$ feet per minute; $p = 184·8$ lb. per square inch (absolute); we have stroke $s = \frac{c}{2n} = 5$ feet 3 inches. Under normal conditions the engine is supposed to work with a high rate of expansion, and to be capable of developing still more power by adjusting the valve gear of the H.P. cylinder. For the normal power developed (6,300 I.H.P.) let a total expansion of $\frac{1}{\epsilon} = 11·1$, $\epsilon = 0·09$, be assumed.

1. *Using in the first instance the shortened method referred to on page* 19.

According to Table No. 3, $\epsilon = 0{\cdot}09$, $\frac{p_t}{p} = 0{\cdot}3065$; therefore $p_t = 0{\cdot}3065 \times 184{\cdot}8 = 56{\cdot}6$ lb. per square inch.

For an efficiency $k = 0{\cdot}85$ (see Table No. 4) the mean pressure (calculating from the L.P. cylinder) will be $p_m = 0{\cdot}58 \times 56{\cdot}6 = 33$ lb. per square inch.

The diameter of the L.P. cylinder is worked out from the equation—

$$\text{I.H.P.} = p_m \times \frac{\pi D^2}{4} \times \frac{c}{33{,}000}$$

whence $A = \frac{6{,}300 \times 33{,}000}{33 \times 786} =$ about 8,000 sq. in.

Therefore $D =$ about 100 inches.

The diameter of the H.P. cylinder is thence obtained by assuming a cut-off for that cylinder.

Assuming $\epsilon_h = 0{\cdot}6$, we have—

$$\frac{\text{Area of the L.P. cylinder}}{\text{Area of the H.P. cylinder}} = m$$

and (from § 6) $\frac{1}{\epsilon} = m \cdot \frac{1}{\epsilon_h}$; therefore $m = 11{\cdot}1 \times 0{\cdot}6 = 6{\cdot}7$.

Area of H.P. cylinder $= \frac{8{,}000}{6{\cdot}7} =$ about 1,190 sq. in.

Diameter of the H.P. cylinder = 39 inches.

The diameter of the intermediate cylinder is obtained by comparing the engine here under consideration with similar engines which have worked satisfactorily.

According to § 14 the proportional volumes of the cylinders will be from about 1 : 3 : 7 to about 1 : 2·4 : 6, &c. The ratios chosen will therefore be as 1 : 2·7 : 6·7. Thus the area of the intermediate cylinder $= 1{,}190 \times 2{\cdot}7 = 3{,}210$ square inches, and the diameter of the intermediate cylinder = 64 inches.

2. *Calculation from the Theoretical Diagram* (page 17).—(Compare the theoretical diagram of an actual engine, A′B′CF′G, Fig. 13, where the cut-off in the H.P. cylinder is 57 °/₀ instead of 60 °/₀.) Assuming the work done, number of revolutions, and piston speed, to be the same as before, the total expansion being taken as = 11·1; cut-off in H.P. cylinder = 60 °/₀; ratio of cylinder-volumes $\frac{\text{H.P.}}{\text{L.P.}}$ as 1 : 6·7; also clearance in H.P. cylinder = 16 °/₀, in L.P. cylinder = 8 °/₀ (see page 14); fall in pressure between boiler and cut-off (cc′) = 35 lb. per square inch (com-

pare Table No. 5). The cut-off will then be $B'C = \frac{(60+16)\times 149{\cdot}3}{184{\cdot}8}$ $= 61{\cdot}4$ °/₀. Total volume after expansion $A'G = (100+8)\times 6{\cdot}7$ °/₀ of the volume of H.P. cylinder $= 724$ °/₀.

The mean pressure in the theoretical diagram can be obtained with a planimeter, or by calculation from Table No. 3. Selecting the latter method, we have—

$$\frac{\text{Initial volume}}{\text{Final volume}} = \frac{B'C}{A'G} = \frac{61{\cdot}4}{724} = 0{\cdot}085.$$

For this cut-off Table No. 3 gives $\frac{p_t}{p} = 0{\cdot}295$; therefore $p_t = 0{\cdot}295 \times 184{\cdot}8$ $= 54{\cdot}5$ lb. per square inch.

According to the other (shorter) method, p_t was $= 56{\cdot}6$ lb. per square inch. By this second method we get rather larger diameters for the cylinders. The process of calculation of these latter is the same as for the previous method.

3. In the *corresponding engine* as actually built the total I.H.P. $= 6{,}320$; $n = 75$; $c = 786$ feet per minute; $s = 5$ feet 3 inches; H.P. cylinder diameter $= 39{\cdot}4$ inches; intermediate cylinder diameter $= 64$ inches; L.P. cylinder diameter $= 102$ inches.

Ratio of the cylinders, $1 : 2{\cdot}66 : 6{\cdot}7$. Cut-off in H.P., 57 °/₀; in intermediate, 55 °/₀; in L.P. cylinder, 50 °/₀. Work done, in H.P., 1,970 I.H.P.; in intermediate, 2,050 I.H.P.; in L.P., 2,300 I.H.P.

Mean pressure in the theoretical diagram $A'B'CF'G = 52{\cdot}5$ lb. per square inch $= p_t$. Mean pressure reduced to area of the L.P. cylinder, in H.P. $= 9{\cdot}95$ lb. per square inch; in intermediate $= 10{\cdot}4$ lb. per square inch; in L.P. cylinder $= 11{\cdot}65$ lb. per square inch. Therefore $p_m = 9{\cdot}95 + 10{\cdot}4 + 11{\cdot}65 = 32$ lb. per square inch.

Therefore the efficiency will be—

$$k = \frac{p_m}{p_t} = \frac{F_1 + F_2 + F_3}{A'B'CF'G} = \frac{32}{52{\cdot}5} = 0{\cdot}609.$$

4. Had the engine had *five cylinders*, with the intermediate cylinder working on the centre crank, and an L.P. and an H.P. cylinder working on each of the two outside cranks, then the calculation (compare the first method) would be as follows:—

Diameter of each H.P. cylinder $= 38{\cdot}5\sqrt{\frac{1}{2}} = 27{\cdot}2$ inches.
Diameter of each L.P. cylinder $= 100\sqrt{\frac{1}{2}} = 70{\cdot}7$ inches.

If the efficiency had been taken at 0·54 instead of 0·58 when using five cylinders, the cylinders would be of somewhat larger diameter than given above.

§ 16. **Receivers.**—Marine engines have no special receivers; firstly, in order to diminish the weight; and secondly, because the exhaust passages, connecting pipes, and valve chests provide sufficient receiver volume. The following are the usual values given to receivers:—Between H.P. and M.P. = 1·8 to 3·8 × volume of H.P. cylinder; between M.P. and L.P. = 1·3 to 2·3 × volume of M.P. cylinder. The effect of the receiver upon the steam distribution can be ascertained by constructing a diagram of volumes (see § 17).

§ 17. **Construction of a Theoretical Indicator Diagram from the Diagram of Volumes.**—If for a given engine the cylinder dimensions, clearance and receiver volumes, steam distribution, and positions of the crank are known, the volume occupied by the steam in its passage through the engine may be graphically determined at any point of the stroke.

Assuming (1) that the steam is expanded and compressed according to the law Pressure × Volume = constant; (2) that, in combining a volume v_1 at a pressure p_1 with another volume v_2 at a pressure p_2, the final pressure obtained is—

$$p = \frac{p_1 v_1 + p_2 v_2}{v_1 + v_2}$$

then the pressure corresponding to the volume at any given point in the stroke may be calculated, and a theoretical indicator diagram drawn.

This kind of diagram is not suitable for calculating the work done by an engine, because the losses by condensation are not taken into account, and the results obtained are too large. But it gives particulars of the steam distribution, fall in pressure, and any peculiarities that may be looked for in the actual indicator diagrams, and it is therefore as well to draw it, when getting out the calculations for a new engine.

Example.—Construction of a "~~combined indicator diagram~~" or "diagram of volumes" for a triple expansion engine. See Figs. 16 to 21, Plate I.

The following data are assumed:—

DATA.

	H.P. Cyl.	Intermediate Cylinder = I.P.	L.P. Cyl.	First Receiver = R_1.	Second Receiver = R_2.
Cylinder diameters	38·58″	64″	100″	—	—
Ratio of volumes	1	2·7	6·7	2·5	4·05
Volumes as plotted in Fig. 16 (inches)	1·97	5·32	13·2	4·92	7·95
Clearance	16 °/₀	11 °/₀	8 °/₀	—	—
Clearance (as plotted), in inches	·315	·585	1·055	—	—
Cut-off	60 °/₀	56 °/₀	54 °/₀	—	—
Cut-off, in inches	1·18	2·97	7·13	—	—
Exhaust lead	20 °/₀	18 °/₀	16 °/₀	—	—
Exhaust lead, in inches	·394	·955	2·11	—	—
Compression	5 °/₀	12 °/₀	12 °/₀	—	—
Compression, in inches	·098	·638	1·58	—	—

Draw a diagram, the abscissæ of which give the volume swept through by the piston, and the ordinates the angles of the crank, calculated from the upper dead point of the H.P. cylinder. If the volume of the H.P. cylinder be taken as 1·97 inches, then the clearance in this cylinder will be $\frac{1{\cdot}97 \times 16}{100} = {\cdot}315$ inch. The volume of the intermediate cylinder $= 2{\cdot}7 \times 1{\cdot}97 = 5{\cdot}32$ inches, and so on.* In order that the diagram may not be unnecessarily long, receiver I is shortened by 3·94 inches, receiver II by 5·91 inches, and the lower part of the L.P. cylinder volume is drawn in on the left of where it ought to be.

It is assumed that the connecting rods are infinitely long, and an ordinary sine curve is used (compare page 59) to represent the path of the piston. The cranks are set at an angle of 120° to each other. In Fig. 21 (Plate I.) the volume of the stroke EF of the H.P. cylinder is indicated for a given angle of the crank (75°). The distribution of the steam at different points of the stroke being drawn, the volume of steam at any given point can be measured from the diagram.

* The diagram, Fig. 16, was originally drawn to the scale volume of H.P. cylinder = 1·97 inches = 50 mm., but has here been reproduced at one-fourth of the actual size.

Determination of Steam Pressures.

Assumed.—Initial pressure in H.P. cylinder = 185 lb. per square inch. Hence, from the drawing, Plate I., the final pressure after expansion in H.P. cylinder will be = 146·5 lb. per square inch (see Fig. 17).

Assumed.—Exhaust pressure from L.P. cylinder to condenser = 2·82 lb. per square inch. Hence the final pressure of compression in L.P. cylinder = 7·11 lb. per square inch (see Fig. 19). The other pressures are deduced from the following equations for the pressures at different periods of the stroke.

I. Combining the pressure and volume of the $H.P._T$ (146·5 lb. per square inch) with the volume $H.P._B + R_1 + I.P._T$,* having the unknown pressure p_1.

Combined pressure—

$$p_2 = \frac{48 \times 146{\cdot}5 + (18 + 125 + 15{\cdot}1)p_1}{206{\cdot}1} = 34{\cdot}1 + 0{\cdot}768p_1.$$

II. Expansion in $H.P._T$ cylinder + $H.P._B + R_1 + I.P._T$ until point of compression in $H.P._B$.

$$\text{Final pressure } p_3 = \frac{206{\cdot}1}{218} \times p_2 = 0{\cdot}945p_2.$$

III. Expansion in $H.P._T + R_1 + I.P._T$ until point of cut-off in $I.P._T$.

$$\text{Final pressure } p_4 = \frac{207{\cdot}5}{267{\cdot}7} \times p_3 = 0{\cdot}733p_2.$$

IV. Compression in $H.P._T + R_1$ till admission begins in $I.P._B$.

$$\text{Final pressure } p_5 = 0{\cdot}733p_2 \times \frac{177{\cdot}7}{145{\cdot}5} = 0{\cdot}895p_2.$$

The pressures in $H.P._T + R_1$ must next be combined with the clearance of $I.P._B$. The slight fall in pressure is neglected; therefore the pressure in $H.P._T + R_1 + I.P._B$ is p_5.

V. Compression in $H.P._T + R_1 + I.P._B$ until exhaust begins in $H.P._B$. This must be at the same pressure as given above, where $H.P._B + R_1 + I.P._T$ was combined with $H.P._T$; therefore—

$$\text{Final pressure } p_1 = \frac{160{\cdot}4}{158} \times p_5 = 0{\cdot}91p_2.$$

---

* As in all cylinders the period of exhaust is greater than the period of compression, the steam sometimes passes from one side of the piston to the other—from the upper (Top) to the under (Bottom). Suffix T denotes top and suffix B denotes bottom.

From Equations I. and V. we get—

$$p_1 = 103 \text{ lb. per square inch.}$$
$$p_2 = 113{\cdot}5 \quad \text{,,} \quad \text{,,}$$

Hence
$$p_3 = 107 \text{ lb. per square inch.}$$
$$p_4 = 83 \quad \text{,,} \quad \text{,,}$$
$$p_5 = 101{\cdot}5 \quad \text{,,} \quad \text{,,}$$

The H.P. diagram (Fig. 17) can now be drawn.

The pressures for the intermediate and L.P. cylinders are obtained in the same way.

In the I.P.$_T$ the pressure at the end of Period III. is $p_4 = 83$ lb. per square inch.

Final pressure after expansion in this cylinder $p_6 = 59{\cdot}5$ lb. per square inch.

VI. Exhaust from I.P.$_T$ to R_2 + I.P.$_B$ + L.P.$_T$ at the unknown pressure p_7.

$$\text{Combined pressure } p_8 = \frac{125{\cdot}7p_6 + 270{\cdot}5p_7}{396{\cdot}2} = 1{\cdot}33 + 0{\cdot}683p_7.$$

VII. Expansion in I.P.$_T$ + I.P.$_B$ + R_2 + L.P.$_T$ until compression in I.P.$_B$.

$$\text{Final pressure } p_9 = p_8 \frac{396{\cdot}2}{403} = 0{\cdot}984p_8.$$

VIII. Expansion in I.P.$_T$ + R_2 + L.P.$_T$ till the cut-off ends in L.P.$_T$.

$$\text{Final pressure } p_{10} = p_9 \frac{371{\cdot}9}{347} = 0{\cdot}669p_8.$$

IX. Compression in I.P.$_T$ + R_2 until L.P.$_B$ opens.

$$\text{Final pressure } p_{11} = p_{10} \frac{137 + 202{\cdot}5}{48{\cdot}2 + 202{\cdot}5} = 0{\cdot}906p_8.$$

X. Combining I.P.$_T$ + R_2 with the clearance of L.P.$_B$ where the final pressure of compression is 7·11 lb. per square inch.

$$\text{Combined pressure } p_{12} = \frac{250{\cdot}7p_{11} + 26{\cdot}8 \times 7{\cdot}11}{277{\cdot}5} = 0{\cdot}818p_8 + 0{\cdot}68.$$

XI. Compression in I.P.$_T$ + R_2 + L.P.$_B$ until exhaust begins in I.P.$_B$. The same final pressure must be obtained as in equation VI., where I.P.$_T$ was combined with I.P.$_B$ + R_2 + L.P.$_T$.

$$\text{Therefore the final pressure } p_7 = \frac{277{\cdot}5}{271{\cdot}5} \cdot p_{12} = 0{\cdot}836p_8 + 0{\cdot}049.$$

From Equations VI. and XI. we get—

$$p_7 = 38{\cdot}6 \text{ lb. per square inch.}$$
$$p_8 = 45 \quad \text{,,} \quad \text{,,}$$

whence
$$p_9 = 44{\cdot}5 \text{ lb. per square inch.}$$
$$p_{10} = 30 \quad \text{,,} \quad \text{,,}$$
$$p_{11} = 41 \quad \text{,,} \quad \text{,,}$$
$$p_{12} = 37{\cdot}6 \quad \text{,,} \quad \text{,,}$$

The intermediate and L.P. cylinder diagrams (Figs. 18 and 19) can now be drawn.

The diagrams thus obtained show the characteristics of actual diagrams, but their mean pressures are naturally much higher than they would be in actual practice.

SECTION II.

THE UTILISATION OF THE STEAM IN THE ENGINE.

§ 18. **The Fundamental Principle of the Mechanical Theory of Heat is that "Heat and Work are equivalent to one another."** It forms the basis for determining the efficiency of the utilisation of steam in the cylinder (1 British Thermal Unit = 772 ft. lb., and 1 calorie = 424 mkg.). In an engine working with a given pressure of admission p_1 and of exhaust p_2, the total work obtained per unit weight of steam is given by the difference between the total heat of admission and that of exhaust.

If the influence of the walls be neglected, it follows that, the higher the temperature of admission, and the lower the temperature of exhaust, the more work will be available per pound of steam. With saturated steam the amount of heat contained in the steam is proportional to its pressure; therefore to obtain the best results our aim should be to work with high admission and low exhaust pressures. In practice the present limit of boiler pressure is about 300 lb. per square inch, and for the exhaust or condenser pressure 0·71 lb. per square inch absolute. As

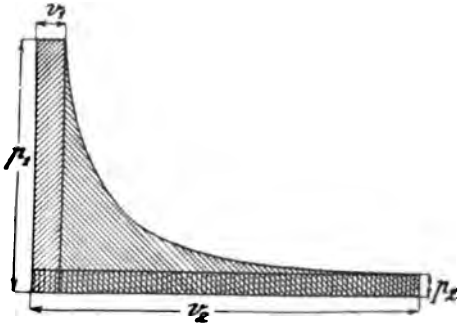


Fig. 22.

all external radiation of heat is a loss, the cylinders and receivers, and all connecting pipes and passages, should be carefully lagged. To add heat

during expansion (that is, to heat the steam after it has entered the cylinder) is not an economical utilisation of heat, because to make the best use of it, it should be added at the highest pressure, *i.e.* during admission. (For the reason why steam jackets are nevertheless useful, see below, § 21.) Hence in a theoretically perfect engine the steam should expand adiabatically,—that is, without heat being either added or subtracted during expansion.

Taking 1 lb. as the unit of weight, and

v_1 = volume of admission in cubic feet (taken from the table of saturated steam, Part VIII.);

v_2 = final volume of expansion (calculated from the equation for adiabatic expansion, see below);

p_1 and p_2 the corresponding pressures of steam (absolute) in pounds per square inch;

we get the following relations (see Fig. 22):—

Work during admission $L_1 = p_1 v_1$ ft. lb.

The equation for adiabatic expansion, according to the mechanical theory of heat, is $p.v^k = \text{constant}$, in which $k = 1{\cdot}135$ for dry steam (see Zeuner's *Technische Thermodynamik*, 1890, vol. ii., p. 75).*

$$\text{Hence } v_2 = v_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{k}}$$

$$\text{Work during expansion } L_2 = \int_{v_1}^{v_2} p \,.\, dv$$

$$= \frac{1}{k-1} (p_1 v_1 - p_2 v_2) \text{ ft. lb.}$$

Work during exhaust $A_3 = p_2 v_2$ ft. lb.

Total work done by 1 *lb. of steam in passing through the engine*—

$$L = L_1 + L_2 - L_3 = p_1 v_1 + \frac{1}{k-1} (p_1 v_1 - p_2 v_2) - p_2 v_2$$

$$= \frac{k}{k-1} (p_1 v_1 - p_2 v_2) \text{ ft. lb.}$$

Table No. 7 is calculated from this equation, and gives the theoretical work in foot pounds for various pressures of admission and exhaust. It furnishes a ready means of ascertaining to what extent an engine is utilising the steam supplied to it.

---

* For low steam pressures this curve varies somewhat from the adiabatic expansion curve.

TABLE No. 7.

Theoretical Work done during the adiabatic expansion of 1 lb. of Steam adiabatically between p_1 *and* p_2, *including the work done during admission and exhaust.*

$$w = \frac{k}{k-1} \cdot (p_1 v_1 - p_2 v_2) \text{ ft. lb. per pound of steam.}$$

[p in pounds per square inch. v in cubic feet per pound.]

$k = 1{\cdot}135$ for dry steam.

$v_1 =$	4·39 c. ft. per lb.	3·01 c. ft. per lb.	2·30 c. ft. per lb.	1·85 c. ft. per lb.
$p_2 =$	$p_1 = 100$ lb. per square inch.	$p_1 = 150$ lb. per square inch.	$p_1 = 200$ lb. per square inch.	$p_1 = 250$ lb. per square inch.
lb. per sq. in.	ft. lb.	ft. lb.	ft. lb.	ft. lb.
1	213,100	233,300	247,900	257,300
3	169,600	192,500	207,700	217,800
6	141,700	163,700	179,200	190,000
9	124,700	146,400	162,250	173,400
12	111,400	133,500	149,400	160,700
15	100,500	123,500	139,000	150,650
20	87,180	109,600	125,900	137,200
25	75,680	98,700	115,000	126,300
30	66,600	89,360	105,800	117,600
35	58,120	81,850	98,450	110,000
40	52,070	74,710	91,300	103,200
45	45,410	68,900	85,120	97,120
50	38,750	62,730	79,560	91,680
60	29,060	53,040	70,360	82,100
70	19,980	45,530	61,510	73,860
80	12,110	36,930	54,010	66,600
90	6,055	30,030	47,710	60,190
100	0	24,100	41,410	54,250

Example.—An engine works with an admission pressure of 150 lb. per square inch absolute, and an exhaust pressure of 3 lb. per square inch absolute. The consumption of feed-water per I.H.P. hour amounts to 13·5 lb. It is required to determine the ratio between the indicated work and the work theoretically available.

$$1 \text{ I.H.P. hour} = 33{,}000 \times 60 = 1{,}980{,}000 \text{ ft. lb.}$$

w according to Table No. 7 is about 192,500 ft. lb. per pound of steam.

Since 13·5 lb. per hour are required, this corresponds to 2,598,750 ft. lb. per hour.

Consequently the work yielded per pound of steam expressed as a percentage of the work theoretically available

$$= \frac{1{,}980{,}000}{2{,}598{,}750} \times 100 = 76{\cdot}2\ \%.$$

With the help of this table it is also possible to ascertain how the steam is utilised in the different cylinders. To determine the work L in each cylinder, the highest pressure shown in the diagram for p_1 and the lowest for p_2 are taken.

In marine engines the comparison is generally expressed as a percentage or efficiency (see § 10), and this is a practical way, provided the engines to be compared are of the same type and size. But if the comparison is extended to engines of different sizes and types, working under different conditions (with wet or dry steam, with or without a condenser, with or without steam jackets, at high or low speeds), this method of comparison by means of the efficiency must be used with great caution, otherwise considerable errors may creep in.

For instance, from Table No. 6 the efficiency of triple-expansion engines is less than that of single-cylinder engines; nevertheless steam is more efficiently utilised in triple-expansion engines than even in compound engines. Hence we see that the efficiency does not afford a reliable standard for comparing the economical utilisation of steam in an engine.

The maximum work obtainable from the steam, calculated according to the equation on page 33, is reduced in practice by losses of various kinds. The reasons for these losses will now be briefly discussed.

§ 19. **Losses by Throttling or Wire Drawing during Admission.**—The cause of this is that marine engines work with a late cut-off, and the valve gear does not permit of a sufficiently large opening for the quantity of steam required to be admitted, nor does it cut off with sufficient rapidity (see Fig. 23). Another reason is that the engine

may have too small a H.P. valve chest, so that the steam expands into the cylinder without being replaced with sufficient rapidity by fresh boiler steam, or the steam pipes may be too small. The loss of work due to this cause is shown by the area of the shaded triangle, *a b c*, in the diagram (Fig. 23).

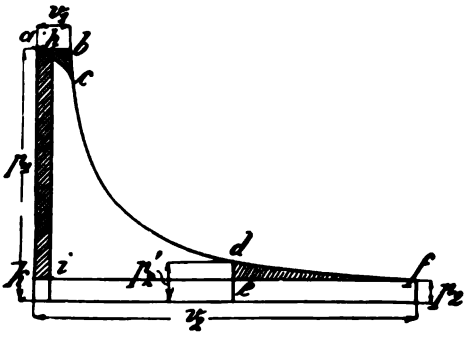


Fig. 23.

§ 20. **Direct Loss of Work due to Clearance.**—At each dead point the piston leaves a space, from the piston to the slide valve, which must be filled with fresh steam at each stroke. (For Clearance, see page 14.) Thus a part of the work of the unit weight of steam (which forms the basis of the diagram), represented by the area *a h i k*, Fig. 23, is lost, *a h* representing the volume of the clearance.

This unavoidable loss of work must not be confused with the loss due to the influence of the cylinder walls (see § 21). To reduce its injurious effect, a portion of the steam is compressed into the clearance, in order that the latter may not have to be filled with entirely fresh steam at each stroke. This does not result in as great an advantage as is usually supposed; for, as shown by Fig. 24, the loss due to incomplete expansion is greater than before, and the gain is merely the difference between

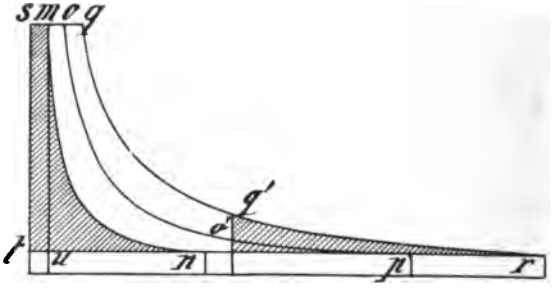


Fig. 24.

the areas *s m u t* and *q' r p o'*. Under certain conditions it may even be a negative advantage, and compression would then be a disadvantage contrasted with the filling of the clearance space with fresh steam.

Since the curves $m\,n$, $o\,p$, $q\,r$, all have the same exponent k, and the distances $s\,m$ and $o\,q$ are the same, the two areas $s\,m\,n\,t$ and $o\,q\,r\,p$ are equal to each other, and therefore the work per unit weight of steam is less with a clearance and compression than without a clearance, because expansion is not complete. Hence to utilise the steam effectively it is usual to make the clearances as small as possible.

§ 21. **Indirect Loss of Work due to the Influence of the Cylinder Walls.**—The question here is not one of losses due to radiation, for the cylinders are always carefully covered with non-conducting materials, in the same way as the pipes. We are dealing with phenomena which take place on the inner side of the cylinder walls, and their action cannot be observed externally.

The temperature of saturated steam varies with its pressure. During one revolution the temperature of the steam will pass through the whole range lying between the temperature corresponding to the limits of pressure; for instance, in a single-cylinder condensing engine it will pass from 338° F. to 140° F., corresponding to 115 lb. per square inch admission and 3 lb. per square inch exhaust pressure. The walls of the cylinder and clearance spaces follow these variations of temperature, as far as they can. This is only possible by the incoming steam giving up some of its heat to the walls of the cylinder and clearance spaces, and this heat is given back by the walls later in the stroke, when the steam is at a lower temperature. In other words, heat is withdrawn from the steam at high pressure, and restored to it at a lower pressure. This occasions a loss of heat, as compared with an engine in which heat is not withdrawn by the walls, when the pressure is highest.

§ 22. **Object of the Steam Jacket.**—It is only the innermost layers of the cylinder walls which participate in all the fluctuations of temperature taking place in the cylinder. The variations of temperature in the outer layers will be less. Each concentric layer has a mean temperature which, diminishing towards the exterior surface of the walls, approximates to the surrounding temperature. The higher the latter, the less far will the variations of temperature extend outwards through the walls, and the smaller will be the exchanges of heat during one revolution. This explains in a few words the real value of the steam jacket.

§ 23. **Influence of Multiple Expansion.**—Expanding the steam in several cylinders has the advantage of diminishing the losses due to the clearance mentioned in § 20. A comparison may be made by means of the diagram, Fig. 25.

The area $a\,b\,r\,s$ represents the diagram of a single-cylinder engine.

Compression is assumed to be up to the initial pressure; the volume of steam admitted per stroke is $a\,b$. Let the cylinder volumes $f\,g$, $h\,i$, $k\,l$, represent the equivalent triple expansion engine, the steam being com-

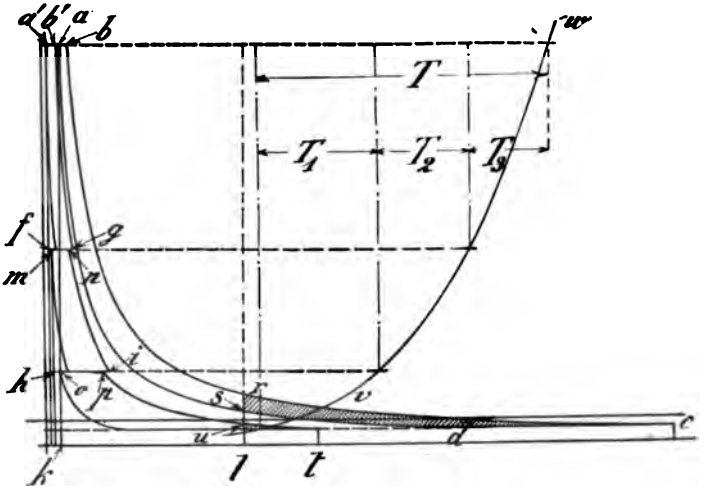


Fig. 25.

pressed in each cylinder to the initial pressure, and the admission volume $a'\,b' = a\,b$, the admission volume for cylinder II will then be $m\,n$ = the exhaust volume of cylinder I. In the same way the admission volume for cylinder III is $o\,p$ = the exhaust volume of cylinder II. As all the curves have the same equation, it is clear that the advantage of multiple expansion in this case is equal to the difference between the areas $c\,r\,s\,d$ and $u\,l\,t$.

Dividing the expansion between several cylinders has the further advantage of reducing the so-called "initial" condensation, by reducing the temperature range in each cylinder. $v\,w$ represents the temperature curve for saturated steam. For a single-cylinder engine the fluctuations would, in the example chosen, be equal to T. By dividing up the expansion they are much reduced for each cylinder, T_3, T_2, T_1. It must be noted that the temperature range in cylinder I is much less than that, for example, in cylinder III.

To divide the total range of temperature into three equal parts would be a mistake, not only because it would give unsuitable areas in the diagrams, but also because steam is more susceptible to cooling at high pressures than at low. In multiple expansion engines the cooling surfaces of the receivers have to be added to those of the cylinders, and the former, as a result of the variations of temperature (not of radia-

tion, which is assumed to be wholly prevented), affect the process of cooling in a similar way to the cylinder walls (see § 21).

§ 24. **Heating the Receivers.**—The transferences of heat from the steam to the walls, and *vice versâ*, mentioned in § 21, may be considerably reduced, if the heat capacity of the walls be diminished. This is best done by heating them. As regards the exchanges of heat, we must note that this heating is not intended to heat the steam itself, but to produce by external action a storing up of heat in the walls, the object being to limit the exchanges of heat between the steam and the walls to as thin a stratum as possible, reckoning from within outwards, *i.e.*, to reduce the quantities of heat participating in the exchange.

In heating the receivers, it must be especially noted that to do so by means of a system of pipes heated with boiler steam is wrong in principle. Either the heat is transmitted from the heating steam to the working steam through the metal (and this, according to § 18, is not so effective as if the heating steam were actually used in the cylinder), or if this direct transference of heat does not take place, the metal walls of the pipes would only increase the surfaces affected by the fluctuations in temperature of the steam, and thus augment instead of diminishing the quantities of heat transferred.

§ 25. **The Condenser.**—According to § 18, the lower the exhaust pressure, the better is the utilisation of the heat supplied to the engine. A considerable improvement can be effected by arranging a special apparatus into which the engine exhausts (see Fig. 26), as compared with an engine exhausting into the open air.

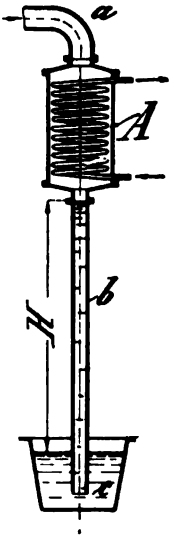


Fig. 26.

The exhaust pipe *a* is led into a vessel A which by an outlet pipe *b* is so connected to an open overflow tank *c* so that the total fall H is more than 30 feet. The vessel A is kept at a constant low temperature by a cooling coil; the exhaust steam is rapidly condensed, and falls to a pressure corresponding to the temperature in A. According to Table XXII., Part VIII., a temperature of 144° F. in the condenser will give a pressure of 1 lb. per square inch absolute. The steam thus condensed to water runs off through the outlet *b*, which, through the pressure of the external atmosphere, is always full of water up to the level H. The apparatus in this shape is unsuited to marine purposes, and must be modified to meet the conditions existing

in vessels. This is effected by substituting for the waste pipe a suction pump, which lifts a quantity of water corresponding to the steam condensed. So far the presence of air has not been considered. If the pump worked too slowly and pumped too little water, the vessel A would fill slowly, the action of the cooling coil would be affected, and the vacuum would become defective. If the pump sucked too much water the condenser would be pumped dry, and a mixture of steam and water would be pumped out. In the first case the apparatus would be practically useless, in the second the pump would have to be forced. The ideal conditions would be reached when exactly as much water was pumped as steam condensed. The gain in work by the use of such an apparatus, as compared with an ordinary non-condensing engine, would be equal to the difference between that part of the area of the diagram of steam lying below the atmospheric line, and the pump diagram. As the upper and lower limits of pressure for the areas of both diagrams are the same, the quantities of work will be directly proportional to the volumes. The ratio of these volumes for equal weights of steam and water is about 1,700 : 1 ; in other words, the loss of work due to the pump, as compared to the gain in work by the use of a condenser, is a negligible quantity.

In actual engines air always finds its way from the feed water, and through leakages in the stuffing boxes, flanges, &c., into the condenser, and reduces the vacuum. Therefore the pump must be of such ample dimensions that it can draw off the air as well as the condensed steam, hence it is usually termed an *air pump*. Theoretical formulæ for the dimensions of the air pump cylinder cannot be given ; in designing new plants, experience alone can serve as a guide. In practical work all risk of leakages must be particularly avoided, and attended to at once if they occur. Particular attention should be paid to the arrangement of the so-called *snifting-valves* (see Pumps). These should only be used in cases of necessity to deaden the shock of the water (if the pump is working very rapidly). Wherever they have to be in constant use there is something defective in the engine.

If the cooling surfaces, quantity of condensing water, &c., are carefully determined from results of actual experience (see Part II., Air Pumps), and there is no leakage, a vacuum of 90 to 95 % of the perfect vacuum may be obtained.

SECTION III.

STROKE OF PISTON—NUMBER OF REVOLUTIONS—TURNING MOMENT—BALANCING OF THE MOVING PARTS.

§ 26. **Stroke, Number of Revolutions, and Piston Speed.**—By piston speed we understand the mean speed of the piston during one stroke. This is—

$$c = \frac{2s \cdot n}{60} \text{ or } = \frac{s \cdot n}{30} \text{ feet per second.}$$

For ordinary values of n, s, and c, see Table No. 8.

TABLE No. 8.

Number of Revolutions, Stroke, and Piston Speed.

Type of Engine.	n	s	c
		inches.	feet per sec.
Torpedo-boats and torpedo-boat destroyers	300 to 400	15 to 20	16 to 19·5
Pinnaces, ships' boats	250 „ 380	6 „ 8	5 „ 8
Small tugs	180 „ 250	8 „ 12	5 „ 8
Small passenger steamers	150 „ 200	11 „ 20	6·5 „ 10
Large tugs and steam trawlers	100 „ 160	12 „ 28	6·5 „ 11·5
Unarmoured cruisers	120 „ 180	24 „ 36	11·5 „ 16
Armoured cruisers	100 „ 150	36 „ 42	13 „ 16
Ironclads	100 „ 120	36 „ 50	13 „ 14·7
Fast steamers	75 „ 95	60 „ 72	13 „ 15·7
Large cargo and passenger steamers	70 „ 90	50 „ 60	11·5 „ 14·7
Small cargo boats	95 „ 130	26 „ 36	10 „ 12·5
Large cargo boats	70 „ 85	35 „ 54	11·5 „ 13

For further details of stroke and number of revolutions, see following tables:—

§ 27. Tables of Particulars of Vessels.

TABLE
Compound

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Engines in the Ship.
Ems*	Fast steamer	Built by Elder of Glasgow for the North German Lloyd. Speed, 16·4 knots.	1 × 6000
City of Chester*	Mail steamer	Inman Line. 1885. 18½ knots. (Busley, Schiffsmaschine.)	1 × 4600
—	Ice-breaker	Built 1895.	1 × 1600
Netherlands	Passenger steamer	Hoboken Ferry. Engines by Fletcher. 12·4 knots. *Engineering*, i., 1894, p. 224.	1 × 740
—	Small passenger steamer	For river traffic.	1 × 180
—	Small passenger steamer	For river traffic.	1 × 125
—	Steam trawler	Built at Vulcan Works, Stettin, 1895.	1 × 200
—	Pinnace	For an armoured cruiser.	1 × 50
—	Ice-breaker	—	1 × 350

* The engines of these ships are only interesting historically, as such large compound engines are no longer made.

No. 9.

Engines.

n No. of Revs.	Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders.				Stroke.		Ratio of Cylinder Volumes.	Total Expansion for a given Cut-off in H.P. Cylinder.
		H.P.		L.P.					
	lb. per sq. in.	ft.	in.	ft.	in.	ft.	in.		Cut-off.
—	95	5	$1\frac{3}{4}$	2 × 7	$3\frac{3}{4}$	5	0	1 : 4	8 for 50 °/₀
—	78	5	8	10	0	6	$5\frac{3}{4}$	1 : 3·1	10 „ 31 °/₀
95	114	2	5	5	1·4	3	2	1 : 4·4	7·3 „ 60 °/₀
119	100	1	6	3	2	2	4	1 : 4·5	7·4 „ 60 °/₀
150	100	1	2	2	1	1	$4\frac{1}{2}$	1 : 3·4	5·7 „ 60 °/₀
175	100	0	11·8	1	8·8	1	2	1 : 3·1	4·4 „ 70 °/₀
130	114	1	3	2	$2\frac{1}{4}$	1	$7\frac{3}{4}$	1 : 3·2	8 „ 40 °/₀
380	142	0	$6\frac{1}{4}$	0	$11\frac{3}{4}$	0	$7\frac{3}{4}$	1 : 3·5	5 „ 70 °/₀
120	106	1	$6\frac{1}{2}$	3	2	1	$7\frac{3}{4}$	1 : 4·5	7·5 „ 60 °/₀

TABLE No. 10. *Triple-Expansion Engi*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Engines in the Ship.
Poseidon	Scouting steamer	Built 1901 by the Bremer Vulcan Vegesak for the German Home Office.	2 × 240
—	Flat-bottomed gunboat	Built by Thornycroft & Co.	2 × 260
Santa Fé	Destroyer	Built by Yarrow for the Argentine Navy. *Engineering*, ii., 1896, p. 122.	2 × 2000
Sword-Fish	Destroyer	Built by Armstrong for the British Navy. Engines by Belliss. *Engineering*, ii., 1896, p. 122.	2 × 2200
Hart Handy Hunter	Destroyer	Built by Fairfield Co. for British Navy, 1896. *Engineering*, i., 1896, p. 245.	2 × 2250
Janus Lightning Porcupine	Destroyer	Built by Palmer's Shipbuilding Co. for British Navy. *Engineering*, ii., 1896, p. 142.	2 × 1900
Satellit	Torpedo cruiser	Built by Schichau, Elbing, for Austrian Navy. *Engineering*, i., 1893, p. 346.	2 × 2300
—	Torpedo-boat	Built by Schneider, Creusot, for French Navy. *Engineering*, ii., 1898, p. 257.	1 × 1500

Torpedo-boats and Destroyers.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders.						Stroke.		Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70% in H.P. Cylinder.	Speed of the Ship.
	H.P.		M.P.		L.P.						
lb. per sq. in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.			Knots.
114	1	2	—		2	2	1	8	1 : 3·36	with 60 % 5·6	—
135	1	0	—		1	8	0	11	1 : 2·7	with 60 % 4·5	—
150	1	6	2	2	3	3½	1	6	1 : 2·1 : 4·8	6·8	26½
192	1	6¾	2	3¼	2 × 2	4	1	6	1 : 2·1 : 4·5	6·4	27
210	1	6¼	2	3½	3	6	1	6	1 : 2·27 : 5·3	7·6	27
210	1	6	2	3½	3	6	1	6	1 : 2·34 : 5·4	7·8	27
180	1	9⅝	2	10⅝	4	1⅝	1	7 11/16	1 : 2·55 : 5·24	7·4	22½
213	1	4 11/16	2	0	2	10¼	1	5 11/16	1 : 2·06 : 4·19	5·9	—

TABLE No. 11. *Triple-Expansion Engi*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Engines in the Ship.
S. 42	Torpedo-boat	Built by Germania, Tegel, for Imperial German Navy. *Schiffbau*, i., No. 18, p. 553.	1 × 1440
—	Torpedo-boat	U.S. Navy. *Engineering*, ii., 1898, p. 819.	2 × 1500
Shirakumo and Asashio	Destroyer	Built by Thornycroft & Co. for the Japanese Government.	2 × 3600
Ardent Boxer Bruiser	Destroyer	Built by Thornycroft & Co. for the British Navy.	2 × 2200
—	Torpedo-boat	Built by Thornycroft & Co.	3000
M	Small cruiser	Building for the Imperial German Navy, 1903.	2 × 5000
Bogatyr	Armoured cruiser	Built at Vulcan Works, Stettin, 1901-2, for the Russian Navy.	2 × 9750
Eber	Gunboat	Built at Vulcan Works, Stettin, 1903, for the German Navy.	2 × 650

Torpedo-boats, Destroyers, and Cruisers.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders.						Stroke.		Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70 % in H.P. Cylinder.	Speed of the Ship.
	H.P.		M.P.		L.P.						
lb. per sq. in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.			Knots.
213	1	$7\frac{5}{8}$	2	$4\frac{3}{4}$	2 × 2	6	1	$5\frac{11}{16}$	1 : 2·16 : 4·67	6·7	—
250	1	2	1	10	2 × 2	$1\frac{1}{4}$	1	6	1 : 2·46 : 6·5	9·3	26
240	1	10	2	6	2 × 2	3	1	7	1 : 1·8 : 3·98	5·7	31
210	1	7	2	3	2 × 2	7	1	4	1 : 2·02 : 4·04	5·77	—
220	1	10	2	5	2 × 2	6	1	6	1 : 1·74 : 3·72	5·3	25
214	2	10	4	5·4	6	9·4	2	6	1 : 2·45 : 5·67	8·1	—
266	3	4	5	0	2 × 5	10	3	0	1 : 2·2 : 5·96	8·5	23
186	1	2·5	1	10·4	2	10	1	8	1 : 2·37 : 5·52	7·9	—

TABLE No. 12. *Triple-Expansion Engine*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Eninges in the Ship.	No Re
Minneapolis	Protected cruiser	Built by Cramp, Philadelphia, for U.S. Navy, 1894.	2 × 6800	13
Victoria Luise	Protected cruiser	Built by Weser, Bremen, for the Imperial German Navy. *Schiffbau*, i., No. 18.	3 × 3300	140
Hai-Yung Hai-Shen Hai-Shew	Protected cruiser	Built at Vulcan Works, Stettin, for Chinese Navy, 1897-8.	2 × 3800	178
Arethusa	Torpedo cruiser	Built by Orlando, Leghorn, for the Italian Navy. *Engineering*, ii., 1893, p. 756.	2 × 2200	265
Buenos Ayres	Cruiser	Built by Armstrong, 1895, for Argentine Navy. Engines by Humphrys, Tennant, & Co. *Engineering*, i., 1896, p. 708.	2 × 7000	154
—	Large cruiser	Building 1903 for the Imperial German Navy.	3 × 6333	120
Brooklyn	Armoured cruiser	Built by Cramp, Philadelphia, 1893, for U.S. Navy.	4 × 4500	136
Powerful	Armoured cruiser	Built for British Navy by Naval Construction Co., Barrow, 1895. *Engineering*, ii., 1896, p. 693.	2 × 12,500	114

Cruisers.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders. H. P.	M. P.	L. P.	Stroke.	Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70 % in H. P. Cylinder.	Speed of the Ship.
lb. per sq. in.	ft. in.	ft. in.	ft. in.	ft. in.			Knots.
156	3 6	4 11	7 8	3 6	1 : 1·97 : 4·79	6·9	23
185	2 6·3	3 7·7	2 × 4 3	2 5½	1 : 2·10 : 5·76	8·2	—
185	2 6	3 8½	2 × 4 0	2 3½	1 : 2·3 : 5·06	7·2	20
170	1 11¼	3 0$\frac{3}{16}$	4 6·1	1 6·1	1 : 2·42 : 5·43	7·8	—
155	3 4	5 0	2 × 5 6	3 0	1 : 2·25 : 5·5	7·8	23·2
207	3 0·6	4 8·3	7 6	3 4	1 : 2·37 : 6·12	8·7	—
155	2 8	3 11	6 0	3 6	1 : 2·47 : 5·16	7·4	21·9
In boiler 260 at engine 230	3 9	5 10	2 × 6 4	4 0	1 : 2·42 : 5·70	8·2	21·8

TABLE No. 13. *Triple-Expansion Eng*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Engines in the Ship.
Infanta Maria Theresa	Armoured cruiser	Built for Spanish Navy by Astilleros del Nervion, Bilbao. *Engineering*, i., 1894, p. 806.	2 × 6800
Iowa	Battleship	Built by Cramp, Philadelphia, for U.S. Navy, 1897.	2 × 5900
Renown	Battleship	Built by Maudslay, London, for British Navy. *Engineering*, i., 1896, p. 79.	2 × 6000
Yashima	Battleship	Built by Armstrong for Japanese Navy. Engines by Humphrys. *Engineering*, ii., 1898, p. 830.	2 × 6750
Majestic	Battleship	British Navy. Engines by Barrow Shipbuilding Co., 1895. *Engineering*, ii., 1898, p. 830.	2 × 5700
Formidable	Battleship	British Navy. Engines by Earle, Hull, 1899. *Engineering*, ii., 1898, p. 830.	2 × 7500
Russell	Battleship	New British Battleship. *Engineering*, ii., 1898, p. 830.	2 × 9000
Kaiser Friedrich III.	Battleship	Built at Imperial Dockyard, Wilhelmshaven, 1900, for Imperial German Navy. *Schiffbau*, i., No. 18.	3 × 4870
Preussen	Battleship	Building 1903 for the Imperial German Navy.	3 × 3530

Ironclads.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders. H.P.		M.P.		L.P.		Stroke.		Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70 % in H.P. Cylinder.	Speed of the Ship.
lb. per sq. in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.			Knots.
143	3	6	5	2	7	8	3	10	1 : 2·18 : 4·79	6·8	20·2
160	3	3	4	7	7	1	4	1	1 : 1·98 : 4·75	6·8	17
155	3	4	4	11	7	4	4	3	1 : 2·16 : 4·8	6·9	18
155	3	4	4	11	7	4	3	9	1 : 2·16 : 4·8	6·9	—
155	3	4	4	11	7	4	4	3	1 : 2·18 : 4·8	6·9	17·5
In boiler 300 at engine 250	2	7½	4	3½	7	0	4	3	1 : 2·68 : 7·1	10	18
In boiler 360 at engine 250	2	9½	4	6½	2 × 5	3	4	0	1 : 2·65 : 7·1	10	19
171	2	10·6	4	6·3	7	2	3	0½	1 : 2·49 : 6·24	8·9	—
192	3	0	4	9	7	4	3	3⅜	1 : 2·45 : 5·9	8·5	—

TABLE No. 14. *Triple-Expansion Engi*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represents the number of Engines in the Ship.
Augusta Victoria	Fast steamer	Built for Hamburg-American Line, at Vulcan Works, Stettin, 1889.	2 × 6000
Spree	Fast steamer	Built for North German Lloyd, at Vulcan Works, Stettin, 1890.	1 × 12,750
Campania Lucania	Fast steamer	For Cunard Line, by Fairfield Co., 1893. *Engineering*, i., 1893, p. 480.	2 × 15,000
Trave Saale	Fast steamer	North German Lloyd. New Engines at Vulcan Works, Stettin, 1895-97.	1 × 8700
Fürst Bismarck	Fast steamer	Hamburg-American Line, at Vulcan Works, Stettin, 1891.	2 × 8200
Kaiser Wilhelm der Grosse	Fast steamer	North German Lloyd, at Vulcan Works, Stettin, 1897. *Engineering*, i., 1898, p. 364.	2 × 14,000
Nile	Mail steamer	Royal Mail Steam Packet Co., by J. & G. Thomson. *Engineering*, ii., 1893, p. 370.	1 × 7700
Majestic	Fast steamer	White Star Line. Harland & Wolff, 1890.	2 × 8500
City of Paris	Fast steamer	Inman Line. Thomson, Clydebank, 1889.	2 × 9200
Prinz-Regent Luitpold	Freight and passenger steamer	North German Lloyd, by Schichau, Elbing. *Engineering*, i., 1895, p. 338.	2 × 2800

Fast Steamers.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders.			Stroke.	Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70% in H.P. Cylinder.	Speed of the Ship.
	H.P.	M.P.	L.P.				
lb. per sq. in.	ft. in.	ft. in.	ft. in.	ft. in.			Knots.
150	3 5·3	5 7	8 10	5 3	1 : 2·62 : 6·6	9·4	18 (mean)
156	2 × 3 1½	6 2¾	2 × 8 2½	5 10¾	1 : 2·0 : 6·92	9·9	18½ (mean)
165	2 × 3 1	6 7	2 × 8 2	5 9	1 : 2·28 : 7·01	9·9	21 (mean)
163	3 8	5 10	9 0	6 0	1 : 2·53 : 6·03	8·6	18
156	3 7	5 7	8 10	5 3	1 : 2·38 : 6·02	8·6	19·5
178	4 4	7 5¾	2 × 8 0½	5 8⅞	1 : 3 : 6·9	9·9	22 (mean)
160	3 2	5 0	7 10	5 6	1 : 2·49 : 6·12	8·9	17·25
180	3 7	5 8	9 2	5 0	1 : 2·5 : 6·54	9·3	19 (mean)
150	3 9	5 11	9 5	5 0	1 : 2·49 : 6·3	9	19 (mean)
175	2 4⅓	3 10	5 10·8	3 11	1 : 2·65 : 6·25	8·9	15·5

TABLE No. 15. *Triple-Expansion En*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represent the number of Engines in the Ship.
Krefeld Aachen	Freight steamer	North German Lloyd, at Vulcan Works, Stettin. 1 screw.	1 × 1750
Iberia	Passenger and cargo steamer	Built at Felton Works, Liverpool. 1 screw. *Engineering*, ii., 1893, p. 206.	1 × 4500
Kherson	Passenger and cargo steamer	For the Russian Volunteer Fleet, by Hawthorn Leslie. 2 screws. *Engineering*, ii., 1896, p. 800.	2 × 6650
Stephan	Cable steamer	Built at Vulcan Works, Stettin, for the North German Ocean Cable Works, 1902-3.	2 × 1200
Giralda	Pleasure yacht	By Fairfield Co., Glasgow. 2 screws. *Engineering*, i., 1895, p. 11.	2 × 4250
Columbia Alma	Cross-channel passenger boats	By Thomson, Clydebank. *Engineering*, i., 1895, p. 209.	2 × 1870
Speedy	Pleasure yacht	By Ramage & Ferguson. *Engineering*, ii., 1896, p. 241.	2 × 200
Hermes	Steam trawler	By Hall, Aberdeen. *Engineering*, i, 1894, p. 352	1 × 418
Delaware	Oil steamer	Dunlop & Co. *Engineering*, i., 1894, p. 209.	1 × 2680
Sylvania Carinthia	Cattle steamers	Cunard Line, by London and Glasgow Engineering Co. *Engineering*, ii., 1895, p. 539.	2 × 2725

Various Ships.

Boiler Pressure above Atm.	Diameter of Cylinders. The small Multiple represents the number of Cylinders.						Stroke.		Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70% in H.P. Cylinder.	Speed of the Ship.
	H.P.		M.P.		L.P.						
lb. per sq. in.	ft.	in.	ft.	in.	ft.	in.	ft.	in.			Knots.
178	1	$10\frac{3}{4}$	3	$2\frac{1}{2}$	5	$0\frac{11}{13}$	5	0	1 : 2·85 : 7·05	10·1	13
180	2	9	4	6	7	4	4	1·2	1 : 2·67 : 7·11	10·2	16
In boiler 250 at engine 160	3	0	4	9	7	8	4	6	1 : 2·51 : 6·53	9·3	19·5
185	1	7·6	2	7·5	4	3	3	4	1 : 2·56 : 6·76	9·7	—
170	2	1	3	4	2 × 3	9	2	3	1 : 2·56 : 6·48	9·3	20·9
160	1	7	2	5	2 × 2	$9\frac{1}{2}$	2	6	1 : 2·33 : 6·03	8·9	19·3
180	1	0	1	7	2	6	1	9	1 : 2·51 : 6·24	8·9	13
160	1	0	1	7	2	$8\frac{1}{4}$	2	0	1 : 2·51 : 7·21	10·3	10·5
160	2	3	3	$7\frac{1}{2}$	5	10	4	3	1 : 2·6 : 6·72	9·6	12·3
170	1	$10\frac{1}{2}$	3	$0\frac{1}{2}$	5	0	4	0	1 : 2·63 : 7·11	10·1	15·3

TABLE No. 16. *Quadruple-Expansion E*

Name of Ship.	Type.	Observations.	I.H.P. The small Multiple represent the number of Engines in the Ship.	*n* No. of Revs.
Cushing	Torpedo-boat	Built by Herreshoff, Bristol. 5 cranks. Inst. Am. Soc. Naval Engineers, 1890.	1 × 1720	—
—	Torpedo-boat	By Schichau, Elbing, 1891. Busley, "Entwickelung der Schiffsmaschine."	1 × 1714	320
Northwest	Passenger steamer	For the American Lakes. *The Engineer*, i., 1895, p. 114.	2 × 3500	120
Victoria Luise	Yacht	Hamburg-American Line, by Blohm & Voss, Hamburg, 1900.	2 × 2100	123
St Louis	Fast steamer	By Cramp, Philadelphia. *Engineering*, i., 1895, p. 800.	2 × 8000	85
Deutschland	Fast steamer	Hamburg-American Line, at Vulcan Works, Stettin, 1900. *Engineering*, ii., 1900, p. 662.	2 × 17,000	78
Emperor William II.	Fast steamer	Built at Vulcan Works, Stettin, for North German Lloyd, 1902-3. *Engineering*, ii., 1903.	2 × 20,000	80
Patricia	Passenger and cargo steamer	Hamburg-American Line, at Vulcan Works, Stettin, 1899. *Schiffbau*, 1900, No. 18.	2 × 2800	74
Friedrich der Grosse Königin Luise	Passenger and cargo steamer	North German Lloyd. Vulcan Works, Stettin, 1896.	2 × 3500	75
Kensington	Passenger and cargo steamer	International Navigation Co. J. & G. Thomson, Clydebank. *Engineering*, ii., 1894, p. 199.	2 × 4150	86½
Singapore	Cargo steamer	By Fleming & Fergusson, 1889. Busley.	1 × 1600	—
Fonar	Cargo steamer	By Wigham Richardson, 1889. Busley.	1 × 1690	—

Various Ships.

Diameter of Cylinders. The small Multiple represents the number of Cylinders.						Stroke.	Ratio of Cylinder Volumes.	Total Expansions with Cut-off 70 % in H.P. Cylinder.	Speed of the Ship.
H.P.		1st M.P.		2nd M.P.	L.P.				
ft.	in.	ft.	in.	ft. in.	ft. in.	ft. in.			Knots.
0	11¼	1	4	1 10½	2 × 1 10½	1 3	1 : 2·04 : 4·03 : 8·17	11·7	22·5
1	4·9	2	0	2 9	3 6½	1 6	1 : 2·01 : 3·81 : 6·3	9	—
2	1	3	0	4 3½	6 2	3 6	1 : 2·07 : 4·25 : 8·78	12·5	17·5
1	7	2	3½	3 3	4 9	3 0·2	1 : 2·08 : 4·25 : 8·9	12·7	—
2 × 2	4½	4	7	6 5	2 × 6 5	5 0	1 : 1·86 : 3·66 : 7·3	10·4	19·5
2 × 3	0·61	6	1·6	8 7·9	2 × 8 10·3	6 0·8	1 : 2·1 : 4·16 : 8·45	12	23
2 × 3	1·4	2 × 4	1	2 × 6 3	2 × 9 4	5 10	1 : 1·73 : 4 : 9	for 75 % 12	22½
1	11	2	9½	4 0	5 10	4 7	1 : 2·15 : 4·46 : 9·46	13·5	13·5
2	1	3	2	4 4·3	6 3½	4 7	1 : 2·3 : 4·3 : 9	12·9	15
2	1½	3	1½	4 4½	6 2	4 6	1 : 2·16 : 4·2 : 8·4	12	15·8
2	0	2	6	3 4	5 0	3 6	1 : 1·8 : 2·8 : 6·4	9·1	—
1	9	2	5	3 6	5 0	3 6	1 : 1·9 : 4 : 8·16	11·7	—

§ 28. **The Crank.**—We shall in future use

P to denote the pull or pressure on the piston rod (pounds).
P^1 ,, pull or pressure on the connecting rod (pounds).
K ,, pressure on the guides (pounds per square inch).
T ,, tangential force acting on the crank circle (pounds).
l ,, length of the connecting rod (feet).
r ,, radius of the crank (feet).
$\lambda = \frac{r}{l}$,, ratio of crank radius to length of the connecting rod.
V ,, momentary speed of the piston (feet per second).
W ,, momentary circumferential speed of the crank (feet per second).
ϵ ,, momentary angular velocity (feet per second).
c ,, mean piston speed (feet per second).
a ,, angle formed by the connecting rod and piston rod.
ω ,, angle of the crank at any given point reckoned from the upper dead point.

The following observations hold good for vertical engines :—

From Fig. 27 the *pull or pressure on the connecting rod* is $P^1 = \frac{P}{\cos a}$

Pressure on the guides, $K = P \tan a$.

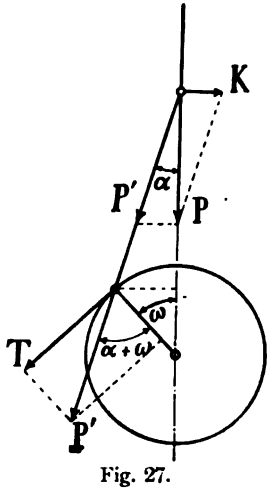


Fig. 27.

K is greatest when the connecting rod is in its most oblique position (assuming that P is constant throughout the stroke). This occurs when $\omega = 90°$. Then $\tan a = \frac{r}{\sqrt{l^2 - r^2}} = \frac{\lambda}{\sqrt{1 - \lambda^2}}$. As $\sqrt{1 - \lambda^2}$ is almost equal to unity, the approximate formula $K_{max} = P \cdot \frac{r}{l} = P \cdot \lambda$. is generally used to calculate the greatest pressure of the guides.

The *tangential force* in the *crank circle* is

$$T = P^1 \sin (a + \omega) = \frac{P}{\cos a} \sin (a + \omega)$$

If the connecting rod be of infinite length, then $a = 0$; therefore $T = P \sin \omega$.

Circumferential Speed and Angular Velocity of the Crank.—The first is understood to mean the arc of the circle traversed in a unit of time. Let $d\omega$ be the angle travelled through in an infinitely short space of time dt, then $rd\omega$ is the

arc of the crank circle passed through in the time dt, and therefore $w = \frac{r d\omega}{dt}$ will be the varying circumferential speed of the crank. The mean circumferential speed is—

$$w_m = \frac{2\pi r \cdot n}{60} = r \cdot \frac{n}{9{\cdot}55}$$

By *angular speed* ϵ is meant the circumferential speed in the circle having a radius equal to unity. This is—

$$\epsilon = \frac{d\omega}{dt}, \quad \epsilon_m = \frac{2\pi n}{60} = \frac{n}{9{\cdot}55}, \quad w_m = \epsilon_m \cdot r$$

Piston Speed.—The table on page 41 gives the usual values for the mean speed of the piston—

$$c = \frac{s \cdot n}{30}$$

Here the value assigned to the variable piston speed v for the angle of rotation of the crank ω must be determined. The travel of the piston for a "connecting rod of finite length" (see Fig. 27) is—

$$x = r(1 - \cos \omega) + l(1 - \cos \alpha).$$

As—

$$l \sin \alpha = r \sin \omega, \text{ or } \sin \alpha = \frac{r}{l} \sin \omega = \lambda \sin \omega,$$

$$\text{then } \cos \alpha = \sqrt{1 - \lambda^2 \sin^2 \omega}.$$

If this be worked out we get—

$$\cos \alpha = 1 - \tfrac{1}{2}\lambda^2 \cdot \sin^2 \omega.$$

and by inserting the values for x we get—

$$x = r(1 - \cos \omega) + \frac{l}{2}\lambda^2 \sin^2 \omega$$

The piston speed at a point situated at a distance x from the upper dead point thus becomes—

$$v = \frac{dx}{dt} = \frac{dx}{d\omega} \times \frac{d\omega}{dt} = \frac{dx}{d\omega} \times \frac{w}{r}$$

Differentiating this we get—

$$v = (r \sin \omega + \tfrac{1}{2} \frac{r^2}{l} 2 \sin \omega \cos \omega)\frac{w}{r} = w(\sin \omega + \tfrac{1}{2}\lambda \sin 2\omega).$$

For "connecting rods of infinite length" $\lambda = 0$ and $v = w \sin \omega$—that is, the piston speed is greatest at the middle of the stroke ($\omega = 90°$) and is equal to the circumferential velocity of the crank for this value of ω, while at the upper and lower dead points it is = 0.

Acceleration of the Piston.—By this expression is meant the change in the piston speed during a given time *dt*, and it is—

$$B = \frac{dv}{dt} = \frac{dv}{d\omega} \times \frac{d\omega}{dt} = \frac{dw}{dt}(\sin\omega + \frac{\lambda}{2}\sin 2\omega) + \frac{w^2}{r}(\cos\omega + \lambda\cos 2\omega).$$

If the circumferential speed w of the crank is uniform, we get $\frac{dw}{dt} = 0$, and therefore—

$$B = \frac{w^2}{r}(\cos\omega + \lambda\cos 2\omega).$$

For "connecting rods of infinite length" $B = \frac{w^2}{r}\cos\omega$—that is, the acceleration of the piston is greatest at the dead points, and is then $= \frac{w^2}{r}$, while it is = 0 in the middle of the stroke.

Acceleration of the Rotating Masses.—The *acceleration of a non-uniformally rotating crank pin* consists of—

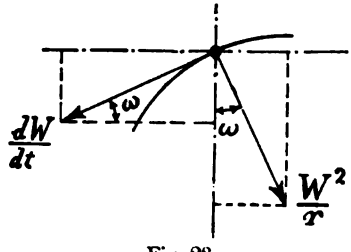


Fig. 28.

Radial acceleration $\frac{w^2}{r}$ and

Tangential acceleration $\frac{dw}{dt}$;

and it may be divided into a vertical and a horizontal component. According to Fig. 28 the vertical component is—

$$\frac{dw}{dt}\sin\omega + \frac{w^2}{r}\cos\omega,$$

and the horizontal component is—

$$\frac{dw}{dt}\cos\omega + \frac{w^2}{r}\sin\omega.$$

If the *rotation* of the crank pin is *uniform* $\frac{dw}{dt} = 0$, because w = constant, and therefore the vertical component will be $\frac{w^2}{r}\cos\omega$, and the horizontal component $\frac{w^2}{r}\sin\omega$.

§ 29. **Moving Parts of the Steam Engine.**—It is convenient to divide these into classes (*a*) those moving *to and fro*, or having a *reciprocating motion*, and (*b*) those having a *rotating motion*. This division gives rise to no inconvenience, except in the case of the connecting rod. In marine engines the ends of these rods, viz., the crank-pin end and the crosshead, are so heavy, that the weight of the rod forms

only a small part of the total weight. We may therefore practically class the head and lower half of the rod with the rotating, and the fork and upper half of the rod with the reciprocating parts of the engine.

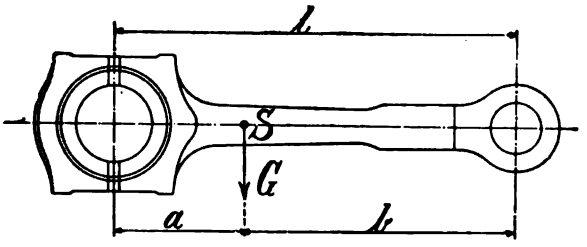


Fig. 29.

The weight of the connecting rod is generally divided up as follows:— Let s be the centre of gravity of the rod, then the weight of the rotating part $= G\frac{b}{l}$, and of the oscillating part $= G\frac{a}{l}$. Therefore, in future, let the ***reciprocating or oscillating parts*** be understood to be the

Piston
Piston rod
Crosshead
Fork of the connecting rod
Rods of the air pump, &c.

$= M_o =$ total mass

and the ***rotating parts*** to be the

Head of the connecting rod
Crank
Crank shaft and shafting
Propeller, &c.

$= M_r =$ total mass

In the same way the masses of the valve gear may be divided into reciprocating and rotating parts, and they will, where necessary, be included under M_o and M_r.

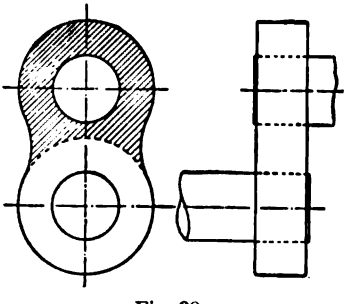


Fig. 30.

Those parts, the centres of gravity of which lie off the centre of the shaft, have a considerable influence in the calculations employed in the balancing of the moving parts. They are called the ***eccentrically rotating parts.*** For instance, the eccentrically rotating parts of the crank

shaft consist of the crank pin and the shaded part of the crank cheeks in Fig. 30. The eccentric rotating parts of an engine are—

(1) The eccentric rotating parts of the crank shaft
(2) Connecting-rod head and end of the crank shaft

$\Big\} = M'_r$

To these we must add the eccentrically rotating part of the valve gear and reversing link, and the lower part of the eccentric rod.

Reduction of the Moving Parts to the Circle described by the Crank.—To simplify the calculations, it is usual to reduce all these masses to the crank circle.

(1.) *Reduction of the Masses relatively to their Radial Acceleration.*—The radial acceleration (centrifugal force) of a mass rotating at a uniform distance r from the centre of the shaft is (see page 60)—

$$\frac{MW^2}{r} = \frac{Mr^2\epsilon^2}{r} = Mr\epsilon^2$$

If it rotates at the distance a, its radial acceleration is—

$$\frac{MW_1^2}{a} = \frac{Ma^2\epsilon^2}{a} = Ma\epsilon^2$$

The radial acceleration of the rotating masses varies as the distances of their centres of gravity from the centre of the crank shaft. In the same way the radial acceleration of the oscillating masses varies as the radii of the cranks upon which they are worked. The reduction is made by multiplying the mass by the ratio of the radius of rotation of the centre of gravity to the radius of the crank.

The reduction of the reciprocating masses to the crank circle is arrived at in the same way. If, for instance, a pump rod with stroke s_1 is driven from a crosshead with stroke s, through a lever, the mass of the pump rod is reduced to the circle described by the crank by multiplying it by $\frac{s_1}{s}$.

(2.) *Reduction of the Masses in relation to their Kinetic Energy.*—The kinetic energy of a rotating mass M is—

$$\frac{M}{2}W^2 = \frac{M}{2}r^2\epsilon^2 = \frac{I_r}{2}\epsilon^2$$

where $I_r = Mr^2$ is the moment of inertia of the mass M at radius r. If the mass rotates at the distance a from the centre of the shaft, its kinetic energy $= \frac{I_a\epsilon^2}{2}$, and I_a will be its moment of inertia for radius a. But

if the mass rotates so that its centre of gravity is in the centre of the shaft, its kinetic energy $= \frac{I \cdot \epsilon^2}{2}$, and I is its polar moment of inertia about that axis.

The reduction of the masses to the circle described by the crank is effected by dividing their moments of inertia by r^2, and the quotient gives the desired result.*

§ 30. **Tangential Pressure on the Crank Pin, and Turning Moment of the Multiple-crank Engine.**—The turning moments of all the cranks upon the shaft of the engine are generally made as uniform as possible during one revolution, in order to diminish the variations in the turning moment on the shaft, and to reduce, as far as possible, any torsional strains that may arise. To determine the conditions necessary for a uniform turning moment, experiments are made on actual engines; in other words, they are tested to determine the energy of gyration in the crank circle.

Graphic Method of Determining the Turning Moment in Engines (for examples see page 65 and Figs. 32-35).—The steam pressure on the piston during one revolution (differences of pressure between the top and bottom) is obtained from indicator diagrams, and is plotted as abscissæ on a base representing the travel of the piston. The radial acceleration or centrifugal force of the reciprocating masses of the connecting rod is added graphically. This latter pressure (see page 60) is

$$= \frac{MW^2}{r} (\cos \omega + \lambda \cos 2\omega),$$

where $M = \frac{G}{g}$ = the mass of the { Piston; Piston rod; Crosshead; Upper part of the connecting rod; Air-pump rods }

reduced to the circle described by the crank.

This simple formula is used to determine the centrifugal force, because to assume that rotation is not uniform, and to take count of frictional resistance, would make the study of the turning moment too complicated. As the vertical and horizontal components of the

* Care must be taken to distinguish between the different kinds of rotation of, for instance, the crank pin and the connecting-rod head. The moment of inertia of the first consists of the product Mr^2 and of the polar moment of inertia round the centre line of the crank pin; that of the connecting-rod head is only equal to the product Mr^2, as it has approximately no rotation round its own centre of gravity.

rotating parts combine to produce a radial force, the vertical components are here neglected.

The weight* of the crank cheeks, which is a positive quantity as they descend and a negative as they rise, must be added to the steam pressure and to the radial acceleration. Thus the total weight to be considered is—

M_g = the weight of the { Piston, Piston rod, Crosshead, The whole of the connecting rod, Air-pump rods, Eccentric portion of the crank shaft }

reduced to the circle described by the crank. The weight is reduced to the crank circle in the same way as for the radial acceleration. The frictional resistance, weight, and radial acceleration of the valve gear are not usually taken into account in these calculations, as their influence upon the turning moment of the engines is small.

If the sum of the steam pressures, centrifugal forces, and weight of the rods (plotted as abscissæ over the travel of the piston) be drawn as a line representing the vertical force P for the different cylinders, then the tangential pressure on the crank pin $T = P\frac{\sin(a+\omega)}{\cos a}$ (see Fig. 27, page 58).

Graphic Determination of T *when* P *is given for any position of the*

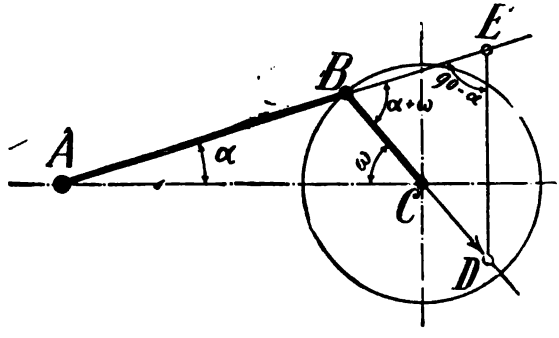


Fig. 31.

Crank (Fig. 31).—Draw AB parallel to the connecting rod, and BC parallel to the crank. Plot the pressure on the piston P from B along BC,

* Strictly speaking, the tangential pressure due to the weight of the crank and the connecting-rod head should be separately estimated, because the path it follows is always a sine curve, and therefore T, constructed as in Fig. 31, is not accurate.

so that BD = P. Draw DE at right angles to AC. Then according to the law of sines DE is the required tangential force T. T must be drawn for each separate cylinder, and plotted over the developed crank circle. The curves of T thus obtained for each cylinder are added together graphically, and the resultant curve represents the total variations in the turning force. The product $T \cdot r$ is the total turning moment of the engine at each instant, during one revolution.

The total turning moment T for one revolution is represented by the area below the curve T between the ordinates AB and CD, Fig. 32. Work of one revolution = area ABCD

$$= \int_0^{2\pi} T r d\omega = T_m \cdot 2\pi r$$

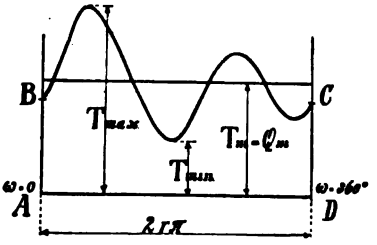


Fig. 32.

Here T_m is the mean turning force on the crank pin, and is found by taking the area ABCD with a planimeter, and dividing the result by $2\pi r$. The value $T_m \cdot r$ is called the *mean turning moment* of an engine. The values $T_{max} \cdot r$ and $T_{min} \cdot r$ are respectively the maximum and minimum turning moments. The mean turning moment, determined graphically, must be equal to that calculated direct from the work shown by the indicator diagrams and the number of revolutions (n), according to the well-known formula—

$$T_m \cdot r = \frac{\text{I.H.P.}}{n} \times 5{,}252 \ (T_m \text{ in pounds, } r \text{ in feet}).$$

As a rule $\frac{T_{max}}{T_{min}} = 1{\cdot}5$ to $2{\cdot}5$, $\frac{T_{max}}{T_m} = 1{\cdot}1$ to $1{\cdot}5$.

The total turning moment is expended in overcoming frictional and propeller resistances.* If the mean value of the sum of these two resistances for one revolution be denoted by Q_m we get the equation $T_m = Q_m$. (For the fluctuations in propeller resistance see "variation in crank-pin velocity," page 69.)

Example.—To determine the tangential pressure of a three-crank engine of about 6,650 I.H.P. Diameter of cylinders, H.P. 3 feet $3\frac{3}{8}$ inches, M.P. 5 feet 4 inches, L.P. 8 feet 5 inches. Stroke, 5 feet 3 inches. $r = 31\frac{1}{2}$ inches ($= 2{\cdot}625$ feet). $n = 75$ revolutions per minute.

* Propeller resistance is the resistance offered by the propeller to turning.

Weight G for H.P. cylinder,	24,861 lb.	Weight M_g for H.P. cylinder,	9,416 lb.
,, ,, M.P. ,,	26,840 ,,	,, ,, M.P. ,,	11,440 ,,
,, ,, L.P. ,,	28,820 ,,	,, ,, L.P. ,,	13,420 ,,

Mean speed of crank pin, $w_m = \frac{2\pi r \cdot n}{60} = 20{\cdot}60$ feet per second. The indicator diagrams are given in Fig. 33.*

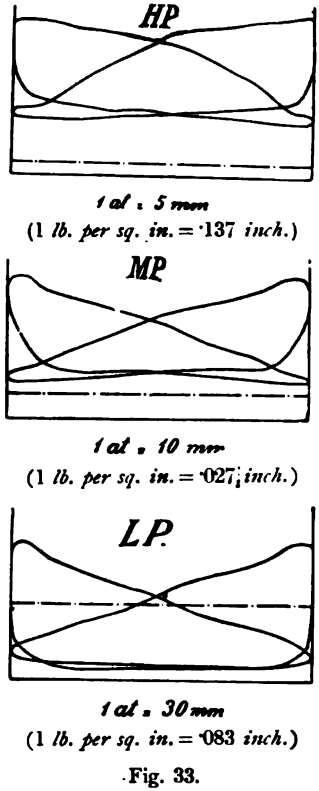


Fig. 33.

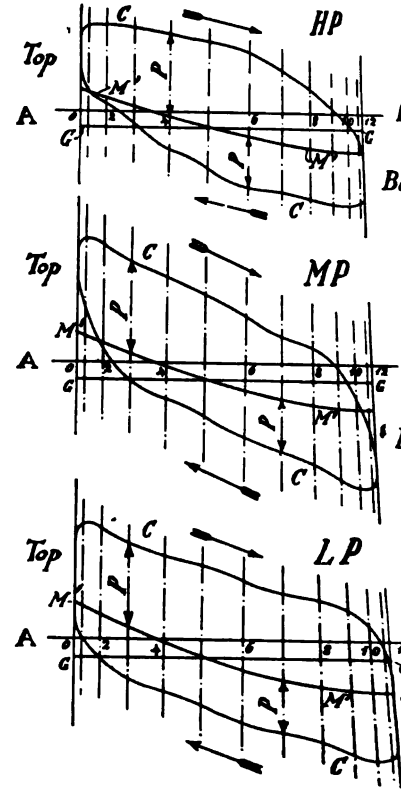


Fig. 34.

Determination of the Vertical Force P (Fig. 34).—All the weight pressures are calculated and reduced to pounds per square inch c area of the L.P. cylinder. cc gives the difference in pressure abov below the piston plotted from the axis AA; GG gives the weights p from AA (H.P. 24,861 lb. = 3·04 lb. per square inch reduced to th

* The scale of the springs refers to the original diagram and has he reduced to one-third.

of L.P. cylinder, &c.). MM gives the accelerating forces due to inertia calculated from the formula $M' = \frac{MW^2}{r}(\cos\omega + \lambda\cos 2\omega)$ with $\lambda = \frac{r}{l} = \frac{1}{4}$,

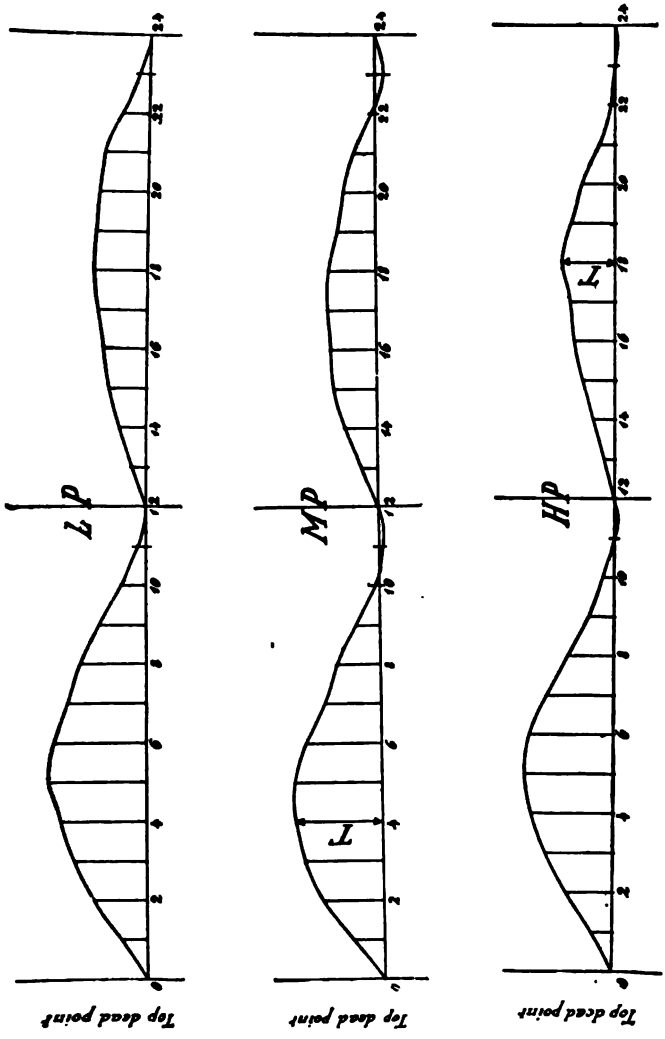


Fig. 35.

plotted from GG. The abscissæ 0, 2, 4, 6, 8, &c., correspond to angles of the crank 0°, 30°, 60°, 90°, 120°, &c., for connecting rods of given lengths. The ordinates between the lines CC and M'M' give the required vertical pressure P. The tangential pressures T for the H.P., M.P., and

L.P. cylinders (Fig. 35) can be determined from the values of P obtained from Fig. 31. These may be combined in the tangential pressure

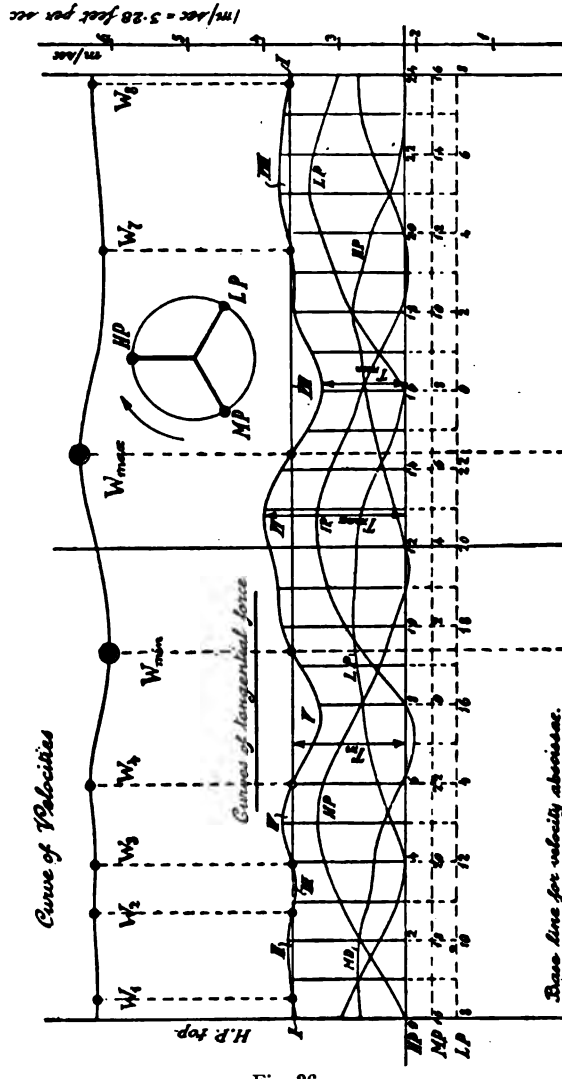


Fig. 36.

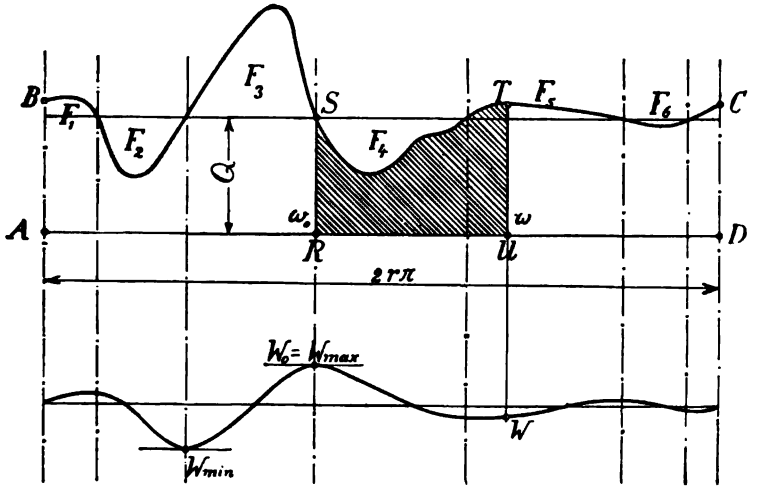
curve (Fig. 36), regard always being had to the proper sequence of the cranks. Hence we have $\frac{T_{max}}{T_{min}} = 1{\cdot}7$, $\frac{T_{max}}{T_m} = 1{\cdot}25$, $T_m = 177{,}400$ lb. T_m may also be calculated direct from the indicated work according to

the formula $T_m = \frac{I.H.P.}{n} \times 5{,}252 \times \frac{1}{r} = \frac{6{,}650}{75} \times 5{,}252 \times \frac{1}{2{\cdot}625}$; whence $T_m =$ 177,400 lb. as above.

§ 31. **Variation in Crank-pin Velocity.**—If w_{max} be the highest, w_{min} the lowest, and w_m the mean value of the velocity of the crank pin during one revolution, the degree of irregularity will be—

$$\delta = \frac{w_{max} - w_{min}}{w_m}$$

Determination of δ.—The resistance of the propeller Q varies very little during one revolution, if the shaft is not subjected to great variations of torsional strain. It may therefore generally be taken as a constant, and $Q = Q_m$. The areas F_1, F_2, F_3..... between the curve T



Figs. 37 and 38.

and the line Q (Fig. 37) thus represent the positive or negative work of the turning force T in relation to Q. This work serves to increase or diminish the kinetic energy of the moving parts. Hence the following equation holds:—Work between ω_0 and ω = variations in the kinetic energy between w_0 and w, or—

$$\frac{M}{2}(w^2 - w^2_0) = \int_{\omega_0}^{\omega} (T - Q) r d\omega = \text{area RSTU} - \overline{RU} \times Q \qquad (1)$$

The curve of crank-pin velocity (Fig. 38) will thus change its direction when the maximum or minimum of the turning force is developed,

and will show a maximum or minimum speed when the curve T cuts the line Q. The greatest crank-pin velocity will be at the point where the algebraical sum of the successive areas $F_1 + F_2 + F_3 + \ldots\ldots$ reaches its highest positive value = X, and the lowest speed where it attains its highest negative value = Y. The change in kinetic energy between its maximum and minimum values corresponds to the difference between the greatest positive and negative areas. Therefore—

$$\frac{M}{2}(w^2_{max} - w^2_{min}) = X - Y \qquad (2)$$

Note that Y must have a negative sign before it. The degree of irregularity agrees with the definition—

$$\delta = \frac{w_{max} - w_{min}}{w_m} \qquad (3)$$

We may take approximately—

$$w_m = \frac{w_{max} + w_{min}}{2} \qquad (4)$$

From Equations (2), (3), and (4) we deduce—

$$M\delta w^2_m = X - Y \qquad (5)$$

δ can be calculated from Equation (5). If δ be known, w_{max} and w_{min} may be found from Equations (3) and (4). The velocity at any given moment (corresponding to the angular velocity ω) can be obtained from Equation (1). In place of an arbitrary speed at starting, w_o, the known speed w_{max} may be substituted. To obtain this—

$$\frac{M}{2}(w^2 - w^2_{max}) = \int_{\omega_{max}}^{\omega} (T - Q) r d\omega = L = \text{difference in the work of}$$

T and Q between the ordinates RS and UT. Hence the

$$\text{speed required } w = \sqrt{w^2_{max} + \frac{2L}{M}} \qquad (6)$$

With the help of this equation the curve of crank-pin velocity may be deduced from the diagram of tangential forces.

By mass M is understood to mean here all the rotating parts reduced to the crank circle, including the screw, shafting, &c. (see pages 60 and 63); also those parts of the reciprocating masses which contribute to the turning moment of the engine. To calculate all these exactly would be too long and complicated a process. We may assume for them the following values:—$\frac{3}{4}$ of the reciprocating mass of one connecting rod, in engines with three cranks at a less angle than 120°; $\frac{4}{4}$ of the reciprocating

mass of one connecting rod, in engines with four cranks at a less angle than 90°, the masses of the connecting rods being the same.

In vessels with long shafts of small diameter, the regulating or steadying effect of the screw does not extend to the engine end of the shafting, and the degree of irregularity is greater near the engine than close to the screw. This irregularity δ is seldom more than 6 % at the screw, and 12 % at the engine. In all cases where great irregularity in running is noticeable, it is caused by the great variations in torsional strains on the shaft. (Compare on this point *Zeitschrift des Vereines Deutscher Ingénieure*, 1902, and *Jahrbuch des Schiffbautechnischen Gesellschaft*, 1903, page 403.) In engines with short, thick shafts (fast steamers, ironclads, ice-breakers) it is the same throughout the whole length of the shaft, and varies between 4 % and 7 %. In ships, therefore, with long, slender shafting, the mass of the screw must not be included in M, when calculating W. If the curve T be regular, and M relatively large, as in ships with a heavy screw and built-up crank shaft, Q may be taken as a constant when calculating w from Equation (6). In other cases the curve of speeds is determined, assuming a constant resistance of the propeller. As this, according to the latest experiments, is approximately proportional to the 3·8 power of the circumferential velocity, $Q = kw^{3\cdot8}$, the value of Q may be found by means of the first approximate value of w, and being thus obtained, the correct curve of speeds can be determined (compare Fig. 39).

§ 32. **Example.**—To determine the mean crank-pin velocity curve, the mean curve of tangential forces given on page 68 must be used.

The ship is supposed to have comparatively short and thick shafting, therefore all the rotating parts may be understood as included under M.

The total mass M reduced to the crank circle is

$$M = \frac{G}{g} = \frac{110{,}000}{32} = \text{about } 3{,}437$$

corresponding to a reduced weight of 110,000 lb.

The areas F_1, F_2, F_3, &c. (marked I, II, &c., in Fig. 36) are:—

	ft. lb.		ft. lb.
$I = F_1 =$	− 1352·5		
$II = F_2 =$	+ 6762·8	$F_1 + F_2$	= + 5410·2
$III = F_3 =$	− 2567·7	$F_1 + F_2 + F_3$	= + 2842·5
$IV = F_4 =$	+ 13019·4	$F_1 + F_2 + F_3 + F_4$	= + 15861·9
$V = F_5 =$	− 58948·9	$F_1 + F_2 + F_3 + F_4 + F_5$ . . .	= − 43086·9 = Y
$VI = F_6 =$	+ 86579·0	$F_1 + F_2$ . . . . + F_6 . .	= + 43492·0 = X
$VII = F_7 =$	− 75367·8	$F_1 + F_2$ . . . . . + F_7 .	= − 31875·8
$VIII = F_8 =$	+ 31875·8	$F_1 + F_2$ . . . . . + F_8 =	0

Therefore $X - Y = 43{,}492{\cdot}0 + 43{,}086{\cdot}9 = 86{,}579$ ft. lb.

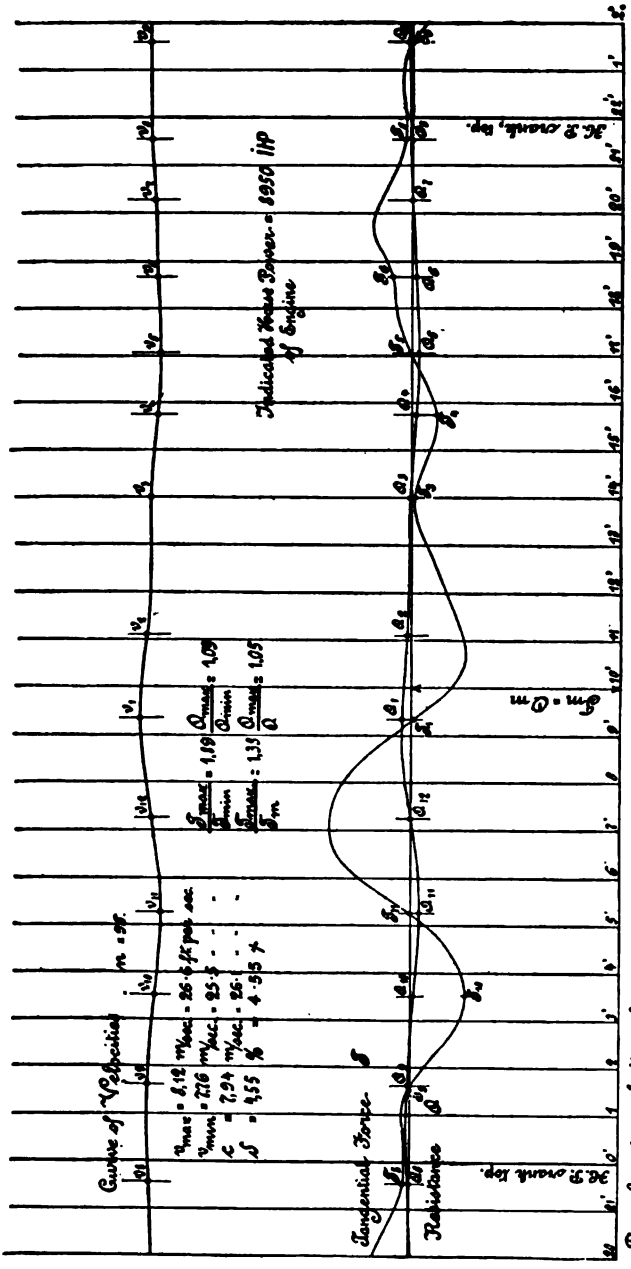

Curve of Velocities
Tangential Force
Resistance
Base line for velocity & force.
Indicated Horse Power = 8950 IHP of Engine
n = 95
Qmax = 1.19 Qmin = 1.09
Qmax = 1.33 Q = 1.05
δm = Qm

Fig. 39.

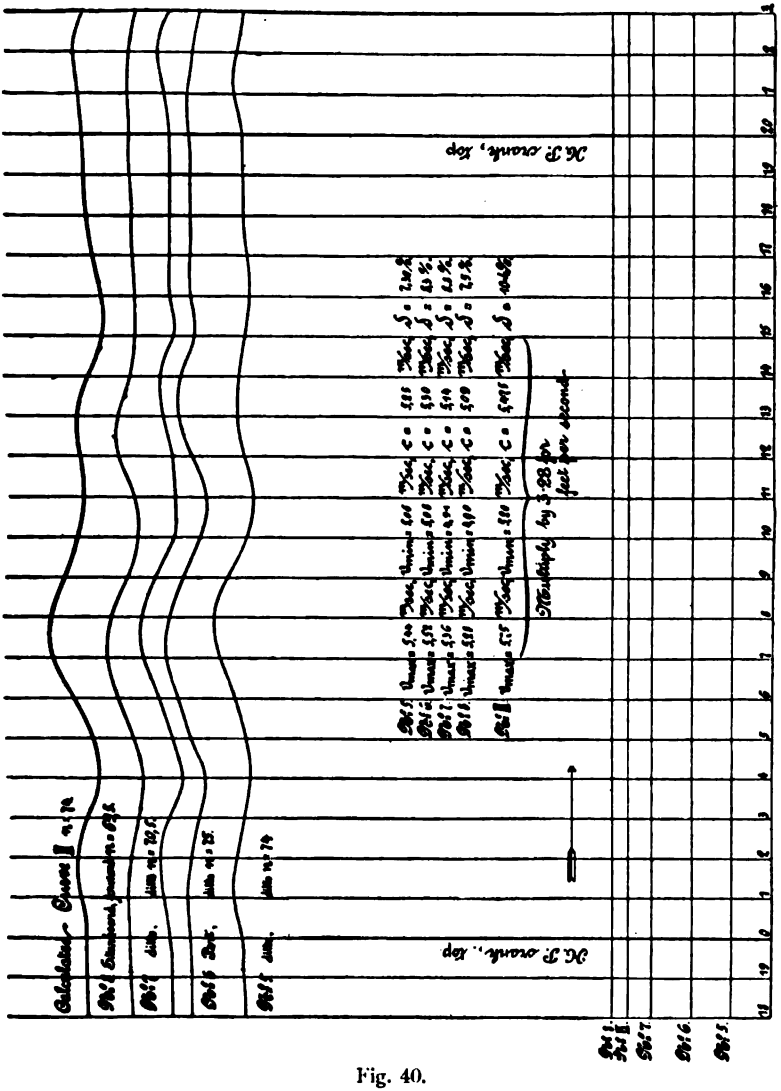


Fig. 40.

δ can now be calculated, and is according to Equation (5)—

$$\delta = \frac{X - Y}{M \cdot w_m^2} = \frac{8{,}6579}{3{,}437 \times 20{\cdot}6^2} = 0{\cdot}059 = 5{\cdot}9\ \%,\ \text{say}\ {\cdot}06\ \text{or}\ 6\ \%.$$

From Equations (3) and (4)—

$$w_{max} = w_m \left(1 + \frac{\delta}{2}\right) = 21{\cdot}2 \text{ feet per second.}$$

$$w_{min} = w_m \left(1 - \frac{\delta}{2}\right) = 20{\cdot}0 \text{ feet per second.}$$

The remaining speeds are obtained from Equation (6), by inserting the areas of work, F_7 (VII), $F_7 + F_8$ (VII + VIII) successively, in place of L. For example—

$$w_7 = \sqrt{21{\cdot}2^2 - \frac{2 \times 75{,}367}{3{,}437}} = 20{\cdot}2 \text{ feet per second.}$$

In this calculation the frictional and propeller resistance Q is assumed to be a constant throughout the revolution. To calculate the speed curve more accurately, the above curve of w should be used to determine the curve of resistance Q according to the formula $Q = k w^{3{\cdot}8}$. The factor k may be approximately obtained from the formula—

$$k = \frac{Q_m}{w_m^{3{\cdot}8}} = \frac{T_m}{w_m^{3{\cdot}8}} *$$

The corrected speed curve is calculated, as before, from the positive and negative areas between the curves of T and Q.

§ 33. **Explanation of Figures.**—Fig. 39 shows (for a fast steamer with two four-crank engines, balanced on the "Schlick" principle) the curves of turning moment and of the calculated resistances Q, as well as the speed curve, calculated from various approximations. In Figs. 39 and 40 the crank-pin velocities are denoted by v instead of w, and the mean of these speeds by c. Fig. 40 shows how the calculated speed curves and those obtained by experiments agree, in a cargo and passenger steamer with two four-crank balanced engines. Curve No. II. has been calculated as described; curves 5, 6, 7, and 8 are taken from experiments.†

§ 34. **Variations in Torsional Strains of the Shafting.**—The points considered in §§ 31 to 33 hold, on the assumption that the shaft, together with the masses of the crank shaft and of the propeller,

* The value for the exponent of $w_m^{3{\cdot}8}$ requires confirmation by further experiments.

† For further details see *Journal of the Schiffbautechnischen Gesellschaft*, vol. i., 1899, p. 311, &c.

is not subject to varying torsional strains. By this term is understood oscillations which succeed each other in the same way as in the balance wheel of a watch; the masses of the crank and propeller respectively playing the part of the escapement, the elastic shafting, subjected to torsional strains, representing the hair-spring.

Assuming the engine to be stationary, the propeller fixed, and a turning moment $P \cdot r$ to be transmitted from the engine to the shaft, the latter will first twist, *i.e.*, the cranks will turn through a certain small angle relatively to the fixed propeller. This angle may be calculated as follows:—

$$\text{In circular measure } \phi = \frac{P \cdot r \cdot L}{G \cdot I} \qquad \text{or in degrees } \phi = \frac{P \cdot r \cdot L}{G \cdot I} \cdot \frac{180}{\pi}$$

$P \cdot r$ being the turning moment in inch-pounds, G the modulus of elasticity of the shaft in pounds per square inch, I the polar moment of inertia of the section of the shaft in inches^4. If $P \cdot r$ be suddenly released, the shaft will fly back, and the crank-shaft masses will then oscillate, setting up torsional strains about the centre of gravity.

As the elastic force increases in proportion to the deflections from the central position, the oscillations follow the laws regulating the swing of an ordinary pendulum. The *duration* of *one* complete *oscillation to and fro* is thus—

$$T = 2\pi\sqrt{\frac{m}{K}}$$

where m is the mass of the crank reduced to the crank circle, and K is the force which must be developed at the crank circle to produce a twist amounting to 1 inch of arc, measured on the crank circle.

K is determined from the formula for ϕ in the following way:—

$$\phi \cdot r = \frac{K \cdot r \cdot L}{G \cdot I} \cdot r = 1 \text{ inch, whence } K = \frac{G \cdot I}{r^2 \cdot L} \text{ in pounds,}$$

r being the radius of the crank in inches.

If these values are used, we get—

$$T = 2\pi\sqrt{\frac{m \cdot r^2 \cdot L}{G \cdot I}} = 2\pi r\sqrt{\frac{m \cdot L}{G \cdot I}}$$

For practical use it is better to calculate the number of oscillations to and fro per minute, namely—

$$n'_\epsilon = \frac{60}{T} = \frac{30}{\pi r}\sqrt{\frac{G \cdot I}{m \cdot L}}$$

Such *specific oscillations* of the shaft would be produced if the propeller were fixed.

The number of specific oscillations may be calculated in the same way, if the propeller is not fixed. According to the law of action and reaction, a backward swing of the propeller masses must correspond to a forward swing of the crank masses, and *vice versâ*. As the front part of the shaft moves in the same direction as the crank masses, and the after part in the same direction as the propeller masses, there must, at some place between the two, be a node or point of no movement. Its position is determined by the condition that, if the external forces cease to act, the opposing momenta of the propeller and crank masses, reduced to the crank circle, must, at every instant, be equal to each other.

The momentum of each = mass × speed. But the speed of the oscillating masses is proportional to the crank radius r, and to the distance of their fixed points from the node. For it is easy to see that

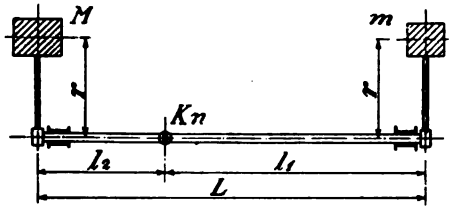


Fig. 41.

from the node, both towards the cranks and the propeller, the amplitude of the oscillations, and therefore the speed of the oscillations, must increase uniformly.

The momentum of each being the same, we have—

$$mv = \text{M}v$$

or by substitution $m(kl_1) = \text{M}(kl_2)$, where—

l_1 is the distance of the node from the centre crank.
l_2 „ „ „ from the propeller.
k is a varying factor.

It follows that—

$$\frac{l_1}{l_2} = \frac{k \cdot \text{M}}{k \cdot m} = \frac{\text{M}}{m}$$

that is, the lengths of the shaft up to the node are inversely proportional

to the masses at either end, reduced to the crank circle. The following is obtained by a simple transposition—

$$\frac{l_1+l_2}{l_1}=\frac{\text{L}}{l_1}=\frac{m+\text{M}}{\text{M}},\ \text{whence}\ l_1=\text{L}\,.\,\frac{\text{M}}{m+\text{M}},\ \text{and therefore}\ l_2=\text{L}\,.\,\frac{m}{m+\text{M}}$$

Thus the position of the node depends wholly on the ratio of the masses m and M, reduced to the crank circle. If $m=\text{M}$, it will be in the centre; if not, it will be nearer to the larger of the masses.

The number of oscillations per minute, in the case of a freely-working propeller, is most easily obtained by deducing it from the formula for n_ϵ already given. In the first case the propeller, and in the second case the node, is stationary; therefore, instead of L, the lengths l_1 and l_2 respectively must be inserted, and thus for the forward end of the shaft we get—

$$n_\text{k}=\frac{30}{\pi r}\cdot\sqrt{\frac{\text{G}\,.\,\text{I}}{m\,.\,l_1}}$$

or, if the value $\text{L}\,.\,\frac{\text{M}}{m+\text{M}}$ be substituted for l_1—

$$n_\text{k}=\frac{30}{\pi r}\sqrt{\frac{\text{G}\,.\,\text{I}\,.\,(m+\text{M})}{\text{L}\,.\,m\,.\,\text{M}}}$$

and the same for the after end of the shaft—

$$n_\text{p}=\frac{30}{\pi r}\sqrt{\frac{\text{G}\,.\,\text{I}}{\text{M}\,.\,l_2}}=\frac{30}{\pi r}\sqrt{\frac{\text{G}\,.\,\text{I}\,.\,(m+\text{M})}{\text{L}\,.\,m\,.\,\text{M}}}$$

Hence $n_\text{k}=n_\text{p}$; *i.e.*, the number of oscillations per minute of the crank masses (n_k) is the same as those of the propeller masses (n_p). The oscillations thus described are the actual torsional oscillations of the shaft, and would be produced if no external forces, especially if no friction or other retarding force, acted upon the oscillating parts. The propeller and crank-shaft masses oscillate in opposite and equal phases, *i.e.*, they reach their middle positions simultaneously, but swing in opposite directions.

On account of the unavoidable friction, and the impeding action of the water on the propeller, the oscillations produced would rapidly become smaller, and ultimately cease. But if there is an *external oscillatory force*, which tends to produce oscillations, these may continue for any length of time. If the oscillations are of a certain amplitude, a kind of equilibrium is established, because the external force does as much work on the shaft, as is annulled by the impeding action of the friction, &c. These two counteracting forces cause the node itself to vibrate and describe a sine curve, but the vibration of the latter compared with the vibrations of the crank and propeller masses is always very small.

The amplitude of the oscillation of the propeller and cranks is, as in

all artificially produced oscillations, proportional to the maximum value of the producing force, and inversely proportional to the impeding forces, and is further dependent on the variations per minute in the producing force. If these are considerably larger or smaller than the periodic oscillations $n_k = n_p$ of the shaft, a strong producing force may only produce small torsional deflections of the oscillating masses. If the recurrent action of this force be nearly equal to the periodic oscillations of the shaft $n_k = n_p$, the amplitude of the torsional deflections of the propeller and crank rapidly increase, until they reach their maximum value; this corresponds to the *critical number of oscillations*, which is attained when the recurrent action of the force is $= n_k = n_p$.

To determine the *torsional oscillations produced in actual working*, let it be assumed that all the oscillating parts—viz., the masses of the crank, shaft, and propeller—are made to rotate at the same number of revolutions n as the engine, while they are subjected to the above-described retarding forces and artificially produced oscillations. As a result of the uneven forces acting on the cranks and propeller, these move very unevenly and jerkily; under certain conditions the piston rods may even appear to stand still at a certain point during each stroke, though the engine continues running.

The *forces* which produce torsional deflection when the engine is at work are the variations which occur in the tangential force (see page 63). Every tangential diagram, however irregular, may be resolved into a constant mean force and a series of harmonic oscillations, which describe complete cycles for each revolution of the engine 1, 2, 3, 4, ... The above method of resolving the diagram is known as *Harmonic Analysis*. On the methods employed see Fischer-Hinnen, *Electrotechnische Zeitschrift*, and Lorenz, *Dynamik der Kurbelgetriebe*. The number of oscillations of these harmonic forces is $= n$, $2n$, $3n$, $4n$ . . . . where n is the number of revolutions of the engine. If one of these series of oscillations chances to be equal in number to the periodic oscillations of the shaft $n_k = n_p$, strong torsional oscillations will be produced. The engine then runs at its so-called *critical speed*, and according to whether n, $2n$, $3n$, &c. $= n_k = n_p$, this critical number of revolutions is said to be of the first, second, or third order.

The *critical number of revolutions* of the shaft is independent of the critical number of revolutions during which the vertical or torsional oscillations of the *ship's body* attain their greatest amplitude. The retarding forces at work are the friction of the engine and shafting, and the rapidly increasing resistance of the propeller, when any alteration in speed takes place. On the laws governing this resistance see page 71.

When calculating the number of periodic oscillations $n_k = n_p$ for practical purposes, the following points should be noted:—One-half of

the oscillating masses, and one-third of the mass of the forward part of the shaft, up to the node, is to be included in the crank masses. One-third the reduced masses of the shaft aft of the node is to be added to the propeller masses; and from 25 to 30 % must be added to the propeller masses, to allow for the water entrained by it. The reduction of the masses must be in the ratio of the square of their actual radii to the radius of the crank circle r.

As the shafting consists of lengths of varying diameters, while the formula for $n_p = n_k$ requires the introduction of a definite moment of inertia I, the different shaftings are reduced to the diameter of the smallest, *i.e.*, of the tunnel shafting. The torsional angle must be the same for the reduced as for the actual lengths of the shaft. If l_1 is the actual length, d_1 the actual diameter of the shaft, d_o the diameter of the tunnel shaft, and l the reduced length of shaft, then—

$$\frac{l_1}{d_1^4} = \frac{l}{d_o^4}\ ;\ \text{whence}\ l = l_1 \frac{d_o^4}{d_1^4} = l_1 \left(\frac{d_o}{d_1}\right)^4$$

The reduced value of flanges, thrust shafts, and cranks can only be estimated approximately. The modulus of rigidity for the material of which shafts are usually constructed (Siemens-Martin steel of 25 to 30 tons per square inch tensile strength, and above 20 % elongation) varies very little, and may, according to the latest experiments, be taken at a mean value of—

$$G = 5{,}260 \text{ tons} = 11{,}782{,}400 \text{ lb. per square inch.}$$

Example.—To calculate beforehand the number of torsional oscillations in an engine of 2,000 I.H.P.—

n = about 75 : stroke = 50 inches : r = 25 inches.

From the working drawings of the engine, the polar moment of inertia of the rotating parts of the crank shaft works out at 63,510 lb. ft.^{2}

Therefore the weight of the same reduced to the crank circle

$$= 63{,}510 \left(\frac{12}{25}\right)^2 = 14{,}630 \text{ lb.}$$

To this must be added:—

The three bottom ends of the connecting rods, totalling	= 4,600 „
And half the oscillating masses, including the pump rods (reduced)	= 6,600 „
Therefore crank masses (without the shaft)	= 25,830 lb.

Similarly the polar moment of inertia of the propeller = 188,700 lb. ft.^{2}
Therefore weight of same reduced to the crank

circle - - - $= 188{,}700 \times \left(\frac{12}{25}\right)^2 =$ 43,450 lb.

To which add 25 % for the entrained water - = 10,860 ,,

Therefore propeller masses (without the shaft) = 54,310 lb.

Polar moment of inertia of the whole shafting = 13,740 lb. ft.^{2}

Whence weight of same reduced to the crank circle

$$= 13{,}740 \times \left(\frac{12}{25}\right)^2 = \text{about } 3{,}166 \text{ lb.}$$

To the crank masses must be added

$$\tfrac{1}{3} \times 3{,}166 \times \frac{54{,}310}{54{,}310 + 25{,}830} = \text{about } 715 \text{ lb.}$$

and to the propeller masses must be added

$$\tfrac{1}{3} \times 3{,}166 \times \frac{25{,}830}{25{,}830 + 54{,}310} = \text{about } 340 \text{ lb.}$$

Thus with all additions the crank masses reduced to the crank circle are—

$$m = \frac{25{,}830 + 715}{386} = \frac{26{,}545}{386} = 68{\cdot}7$$

and the corresponding propeller masses

$$\text{M} = \frac{54{,}310 + 340}{386} = \frac{54{,}650}{386} = 141{\cdot}6.$$

Here the acceleration due to gravity is taken at 32·2 ft. sec.^{2} = 386 inches.

Diameter of smallest shaft (tunnel shaft) = 12·5 inches.

Polar moment of inertia of section of the shaft, having a diameter of 12·5 inches

$$= \text{I} = \frac{d^4\pi}{32} = 2{,}396 \text{ in.}^4$$

Length of the shaft reduced to the diameter (12·5 inches) of the tunnel shaft = 1,592 inches.

We have now all the data for calculating the oscillations. These work out at

$$n_k = n_p = \frac{30}{\pi r}\sqrt{\frac{\text{G . I . } (m + \text{M})}{\text{L . } m \text{ . M}}}$$

$$= \frac{30}{\pi \text{ . } 25}\sqrt{\frac{11{,}774{,}160 \times 2{,}396 \times (68{\cdot}7 \times 141{\cdot}6)}{1{,}592 \times 68{\cdot}7 \times 141{\cdot}6}}$$

= about 236 double oscillations per minute.

Therefore the critical number of revolutions of the shaft are—

$$\text{First order} = \frac{236}{1} = 236$$

$$\text{Second ,,} = \frac{236}{2} = 118$$

$$\text{Third ,,} = \frac{236}{3} = 78{\cdot}7$$

$$\text{Fourth ,,} = \frac{236}{4} = 59, \text{ and so on.}$$

As the engine usually runs at about 75 revolutions per minute, it is only orders III and IV of the critical number of revolutions which need be taken into account. If $n = 80$, very strong torsional oscillations would be recorded in actual running.*

* Compare Lorenz, *Dynamik der Kurbelgetriebe;* Gümbel, *Transactions of the Institution of Naval Architects*, 1902; Frahm, *Zeitschrift des Vereines Deutscher Ingénieure*, vol. xxxvi., 1902; Foettinger, *Jahrbuch der Schiffbautechnischen Gesellschaft*, 1903.

Balance of the Moving Parts.

§ 35. **General Remarks.**—The steam admitted to the space between the piston and cylinder cover exerts upon them both the pressure P. This pressure may be divided into the radial acceleration P_1 and the actual pressure on the piston rod P_2, so that $P = P_1 + P_2$. The force P acting upwards on the cover is transferred through the framing of the engine to the bed-plate, which it tends to lift. To balance it there is only the pressure P_2, as the rest of the steam pressure on the piston P_1 is absorbed in accelerating the piston and connecting rods. The force P_2 is transmitted through the rods to the bed-plate. During the down stroke the unused force P_1 is exerted to raise the framing of the engine. In the same way it can be proved that during the up stroke the weight of the rods tends to force the framing of the engine downwards. The horizontal components of the forces seek to move the frame horizontally, in a direction perpendicular to the longitudinal axis of the engine, while the centrifugal force of the crank endeavours to drag the bed-plate along with it.

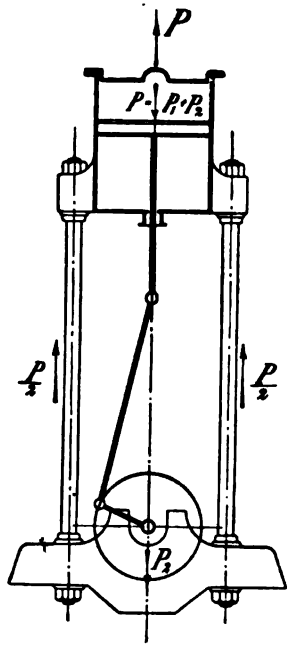


Fig. 42.

In the following considerations it is assumed that—

1. The connecting rod is of infinite length, and that the movement of the reciprocating masses is continuous, similar to that of the crank and the crosshead.

2. The rotation of the crank pin is uniform, and therefore the acceleration of the reciprocating masses follows the law $B = \frac{W^2}{r} \cos \omega$ (see page 60). (ω = angle of the crank calculated from the upper dead point.)

§ 36. **Balancing the Moving Parts of a Single-crank Engine.**—The radial acceleration of the rotating masses can be compensated for by a counterweight on the crank, the weight of which is equal to that of the rotating masses. If the counterweight be made equal in weight to the weight of the rotating and reciprocating masses, the vertical pressure of the latter disappears. A new horizontal force is however introduced, which is equal to the radial acceleration of the reciprocating masses. The reason of this is that the vertical components only of the counterweight, which act upon the reciprocating parts, are absorbed by the latter. By the use of such a counterweight the radial acceleration of the reciprocating parts is deflected through an angle of 90°.

§ 37. **Balancing the Moving Parts of a Two-crank Engine.**—For marine engines, only those which have their cranks at an angle of 90° need be considered, as with any other crank angle the engines work very unevenly.

Let M_1 and M_2 be the reciprocating, and m_1 and m_2 the rotating masses of the H.P. and L.P. cylinders respectively, then the value of the vertical components will be given by the equation—

$$P_1 = (M_1 + m_1)\frac{W^2}{r}\cos\omega \text{ for the H.P. cylinder.}$$

$$P_2 = (M_2 + m_2)\frac{W^2}{r}\cos(\omega \pm 90) \text{ for the L.P. cylinder}$$

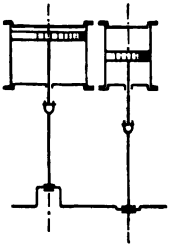


Fig. 43.

(see Fig. 42), the plus and minus values depending on whether the L.P. crank is in advance $(\omega + 90)$ or behind $(\omega - 90)$ the H.P. crank. The value of the *horizontal components* is given by the equation—

$$p_1 = m_1\frac{W^2}{r}\sin\omega \text{ for the H.P. cylinder.}$$

$$p_2 = m_2\frac{W^2}{r}\sin(\omega \pm 90) \text{ for the L.P. cylinder.}$$

The vertical components of the moving parts tend to raise and lower the framing of the engine with a force $P_1 + P_2$, the values of which are shown in Fig. 44 for every angle of the crank ω. At the same time the forces P_1 and P_2 tend to twist the frame in the vertical plane. The horizontal components tend to move the framing in a horizontal direction at right angles to the longitudinal axis of the engine, with a force $p_1 + p_2$, and so to twist it at the same time in the horizontal plane.

The horizontal components can be wholly overcome by balance weights on the cranks m_1 and m_2; if balance weights equivalent to $m_1 + M_1$ and $m_2 + M_2$ be used, the vertical components disappear, but in their place horizontal components appear, namely—

$$P'_1 = M_1 \frac{W^2}{r} \sin(\omega + 180), \text{ and } P'_2 = M_2 \frac{W^2}{r} \sin(\omega + 180 \pm 90).$$

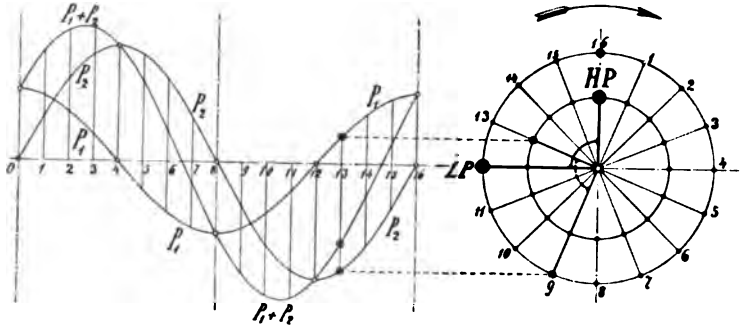


Fig. 44.

§ 38. **Balancing the Moving Parts of a Three-crank Engine.**—As a rule no attempt is made to balance the effect of the reciprocating parts of a three-crank engine, as the components of the moving parts cannot be eliminated without introducing practical difficulties.* In small engines the effect of the rotation can be balanced by corresponding counterweights, as already explained, but this method is not without drawbacks.

In ordinary three-crank engines there still remains a free vertical force, as well as vertical and horizontal moments to be dealt with. As the axis (about which the components of the moving parts tend to turn or tilt the frame) changes its position every instant, the moments above mentioned are usually transferred to the plane of one of the outer cranks (see Reciprocation in a four-crank engine).

If a and b are distances between the cylinders (Fig. 45) α and β, the angles between the cranks (Fig. 46), then the vertical force will be—

$$P = P_1 + P_2 + P_3 = (M_1 + m_1)\frac{W^2}{r}\cos\omega + (M_2 + m_2)\frac{W^2}{r}\cos(\alpha + \omega) +$$

$$(M_3 + m_3)\frac{W^2}{r}\cos(\alpha + \beta + \omega)$$

* Compare Yarrow's suggestion for employing so-called "bob-weights." *Engineering*, 1893, p. 207.

and the horizontal force will be—

$$\mathfrak{P} = \mathfrak{P}_1 + \mathfrak{P}_2 + \mathfrak{P}_3 = \mathfrak{M}_1 \frac{w^2}{r} \sin \omega + \mathfrak{M}_2 \frac{w^2}{r} \sin (\alpha + \omega) + \mathfrak{M}_3 \frac{w^2}{r} \sin (\alpha + \beta + \omega).$$

The vertical components of the moving parts, reduced to plane I

$$= a (M_2 + \mathfrak{M}_2) \frac{w^2}{r} \cos (\alpha + \omega) + (a + b) (M_3 + \mathfrak{M}_3) \frac{w^2}{r} \cos (\alpha + \beta + \omega).$$

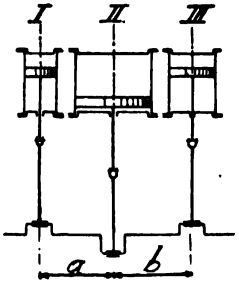


Fig. 45.

The moment of the horizontal component of the moving parts, reduced to plane I

$$= a \mathfrak{M}_2 \frac{w^2}{r} \sin (\alpha + \omega) + (a + b) \mathfrak{M}_3 \frac{w^2}{r} \sin (\alpha + \beta + \omega).$$

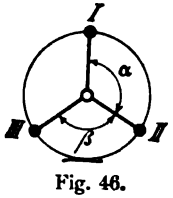


Fig. 46.

These formulæ are for "connecting rods of infinite length," but for connecting rods of finite length cos $\omega + \lambda \cos 2\omega$, &c., must be substituted for cos ω. In ordinary three-crank engines the static moments are very considerable, and if there are no perceptible vibrations in many of these engines it is because the engines are well placed in the ship,* or because the vibrations of the ship do not synchronise with those of the engine (see page 104), or because the masses of the ship's hull are very large in proportion to the moments acting on them.

To secure the balance of the moving parts (apart from considerations of economy of space, weight, &c.), the following is a good arrangement of a three-crank engine:—

1. The L.P. cylinder is placed in the centre, between the H.P. and the M.P. cylinders.
2. The crank angles do not exceed 120°.
3. The cylinders are placed as near together as possible, the valves being placed off the centre-line of the engine.

§ 39. **Balancing the Moving Parts of a Four-crank Engine: Schlick System.**†— Complete balance can be obtained in four-

* See *Zeitschrift des Vereines Deutscher Ingénieure*, 1894, p. 1091. O. Schlick.

† See Schlick, *Transactions of the Inst. of Naval Architects*, 1893, 1894; also Lorenz, *Die Dynamik der Kurbelgetriebe.*

crank engines, on the assumption of infinitely long connecting rods. The columns and bed-plate can be cast in the form of a stiff girder, in order to take up the vertical components P in the vertical plane, and the horizontal components 𝔓 in the horizontal plane. The framing remains at rest if, under the action of the above forces, it is in equilibrium at each instant during a revolution of the engine. This happens, if at each instant—

1. The sum of the vertical components P, and of the horizontal components $\mathfrak{P} = 0$; and

2. The sum of the turning moments of the components P and 𝔓, referred to any plane at right angles to the axis of the shaft, $= 0$. These moments are generally referred to one of the planes of either of the outermost cranks.*

Equilibrium of the engine framing can also be obtained, when the sum of the turning moments about two different planes $= 0$:—if, for instance—

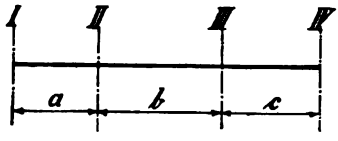


Fig. 47.

1. The sum of the turning moments of P and 𝔓 referred to the plane of the outer right-hand crank $= 0$, and

2. The sum of the turning moments of P and 𝔓 about a plane through the outer left-hand crank $= 0$. These two last conditions are generally used to determine the balance of the moving parts. If we call

a, b, c (Figs. 47 and 48), the distances between the four cranks, that is, the distance between the centres of the cylinders, and α, β, γ the angles between the four cranks—

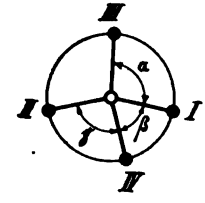


Fig. 48.

M_1, M_2, M_3, M_4† the masses affecting the vertical components of the moving parts P; and

$\mathfrak{M}_1, \mathfrak{M}_2, \mathfrak{M}_3, \mathfrak{M}_4$† the masses affecting the horizontal components of the moving parts 𝔓

* It is a question of the *statical moments* of the components of the moving parts, relative to a given plane. In future, for brevity sake, instead of writing "moment relative to plane, &c.," the usual expression, "moment about plane, &c.," will be used, although of course a force cannot produce rotation about a plane, but only about an axis.

† The symbols M_1, M_2, &c., used here have the same significance as the corresponding letters in § 37, 38.

Then the two equations for the *vertical moments* of the reciprocating parts may be written—

$$(1) \qquad M_2 \cos(\omega+\alpha+\beta+\gamma)\times a + M_3 \cos\omega\times(a+b) + M_4\cos(\omega+\alpha+\beta)\times(a+b+c)=0.$$

$$(2) \qquad M_3\cos\omega\times c + M_2\cos(\omega+\alpha+\beta+\gamma)\times(b+c) + M_1\cos(\omega+\alpha)\times(a+b+c)=0.$$

The equations for the *horizontal moments* of the reciprocating parts are—

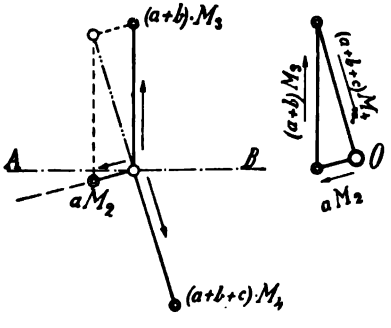
$$(3) \qquad \mathfrak{M}_2 \sin(\omega+\alpha+\beta+\gamma)\times a + \mathfrak{M}_3\sin\omega\,(a+b) + \mathfrak{M}_4\sin(\omega+\alpha+\beta)\times(a+b+c)=0.$$

$$(4) \qquad \mathfrak{M}_3\sin\omega\times c + \mathfrak{M}_2\sin(\omega+\alpha+\beta+\gamma)\times(b+c) + \mathfrak{M}_1\sin(\omega+\alpha)\times(a+b+c)=0.$$

These equations must be determined for every value of ω between 0° and 360°.

The common factor $\frac{W^2}{r}$ disappears, showing that *in a balanced engine the balance is satisfactory for any given crank-pin velocity, and therefore for any number of revolutions, whether running ahead or astern.*

It must be specially noted that, if the above conditions of equilibrium are fulfilled, they will only produce a balance, and prevent vibration in the ship, when the columns and bed-plate of the engine are cast in the form of a stiff girder. In order to make the balance perfect, it is therefore desirable to connect the upper ends of the columns or the cylinders firmly with each other in a longitudinal direction. Light engines should also be braced diagonally in the same direction. If the columns and bed-plate are not sufficiently stiff, they may bend under the weight of the cylinders, and cause vibration in the hull.



Figs. 49 and 50.

In calculating the reciprocation of the moving parts of a given engine, the conditions necessary to secure a good balance apply equally to the rods of the valve gear. In a four-crank engine fitted with Stephenson's link motion there are $4+8=12$ cranks (four cranks and eight eccentrics) which must be balanced. Generally, however, the valve gear is combined into

one resultant factor or figure (see below). This resultant must be further reduced to the crank circle, before combining it with the cranks (see page 62).

§ 40. **To Determine the Balance of the Moving Parts** in actual practice, graphic methods, based upon the four above-named equations, are employed. If the values of aM_2, $(a+b)M_3$, and $(a+b+c)M_4$ are plotted to any scale from the centre along the cranks (compare Figs. 48, 49), the sum of the vertical components must, according to Equation 1, § 39, give $\omega = 0$. It may be proved that such is the case when the polygon formed by the projected moments of aM_2, $(a+b)M_3$ and $(a+b+c)M_4$ on their respective cranks, is closed (see Figs. 49 and 50). The line $(a+b+c)M_4$ must end at the final point 0. If the conditions obtaining in the second equation, § 39, are fulfilled, the polygon made up of the values cM_3, $(b+c)M_2$, and $(a+b+c)M_1$ will be closed. Precisely the same holds for the horizontal balancing of the moving parts. These may be worked out for every angle of rotation ω by drawing a closed polygon with the values of $m_2 a$, $m_3(a+b)$, $m_4(a+b+c)$, and another with the values of $m_3 \cdot c$, $m_2(b+c)$, and $m_1(a+b+c)$, regard being had to the angle of the crank.

A complete vertical and horizontal balance of the moving parts is obtained if all the four polygons, drawn as above, are closed. These polygons are used to determine graphically the positions of the crank, and the weights required to balance the moving parts. The values a, b, and c, from centre to centre of cylinders, the approximate weight of the connecting rods of the two middle cranks, and the angle between them, are generally determined by structural conditions.

For balance in the vertical direction we then get—

1. *Position and weight* M_1 *of crank* I, from the polygon reduced to the plane of crank IV, as given by Equation 2, § 39.

2. *Position and weight* M_4 *of crank* IV, from the polygon reduced to the plane of crank I, as given in Equation 1, § 39.

For the horizontal balance in the same way we get—

3. *Position and weight* m_1 *of crank* I, from the polygon as given by Equation 4, § 39.

4. *Position and weight* m_4 *of crank* IV, from the polygon as given by Equation 3, § 39.

To obtain vertical and horizontal balancing, the positions and weights of the crank must be made to agree (see Example, page 92).

§ 41. **Study of the Valve Gear.**—As we have already said, the moments due to the moving parts of the valve gear are

reduced to one moment about plane IV, and to one about plane I. Let OA, OB, OC, &c. (Fig. 51), be the moments due to the aforesaid pressure of the valve gear relative to plane IV, plotted respectively in the direction of their eccentrics. They may be combined into one polygon of moments as in Fig. 52. Then the line O_1H will be the resultant moment of the eight moments OA, OB, &c. In the same way the moments of the pressure due to the masses of the moving parts of the valve gear may be combined into a polygon relative to plane I. The length of the line closing the polygon gives the value of the resultant moment, and the direction of this line gives the position of the ideal or hypothetical crank.

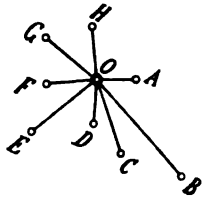


Fig. 51.

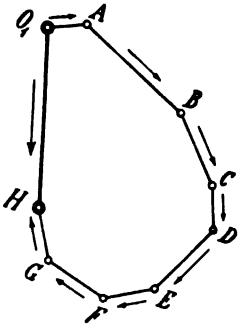


Fig. 52.

To determine the position and mass of crank I for the vertical balancing of the moving parts *with reference to the valve gear*, the lines $M_2(b+c)$, M_3c, and O_1H, starting from C, Fig. 53, must be successively plotted to form a polygon.* For determinating O_1H the moment of the vertical pressure due to the valve-gear masses alone must be used. The closing line CH, produced in the direction of the arrow, gives the moment of pressure due to the masses of crank I, and its direction. To determine the mass of crank I for the vertical balance, divide the moment CH by the distances $a+b+c$. The mass and direction of crank IV for the vertical balance, and of cranks I and IV for the horizontal balance, are obtained in the same way. The masses and directions of the cranks differ somewhat for the vertical and horizontal balances, and a mean of the two should therefore be taken (compare Example II., page 100). The vertical balance is often used alone, without combining it with the horizontal, as it is of more importance in preventing vibration than the latter.

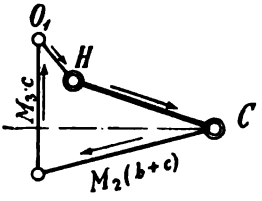


Fig. 53.

§ 42. **Remarks.**—1. If the valve gear is outside one of the end

* The direction of the line O_1H should be the same in Figs. 52 and 53.

cranks, the moment about this crank due to the pressure of its masses must be plotted in the opposite direction in the polygon, as compared with the moments of the valve gear when inside the cranks.

2. If some of the valves are worked by a double-armed lever, the masses on the further side of the fulcrum must be treated as negative.

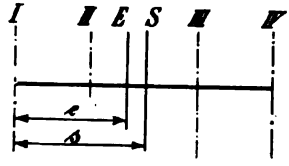


Fig. 54.

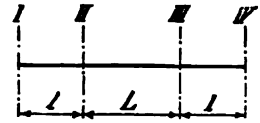


Fig. 55.

3. If parts of the valve gear are worked by an eccentric in different lateral planes—say E and S, Fig. 54—the calculation of the moment of the masses relative to plane I must be made by multiplying the moments in plane E by *e*, those in plane S by *s*. Also, to determine the direction of the two moments in the polygon, the direction of the two actuating eccentrics must be taken into account.

4. When a slide valve is fitted with an auxiliary relief piston, the weight of both must naturally be taken into account in the calculation.

5. If the slide-valve rods or the pump rods are not on the centre line of the valve gear, a special vertical balance of the moving parts about this centre line ought, strictly speaking, to be drawn out for them. This is not usually done, because these masses are always small, and the rods are treated as though they were on the centre line of the valve gear.

§ 43. **Most favourable Arrangement in an Engine to secure Perfect Balance of the Moving Parts.**—The equations in § 39 show that balancing is most effectually carried out when—

1. The cylinders with the heaviest connecting rods are in the centre.

2. The distance between the two middle cylinders is as large as possible, as compared with the distances between the two outside cylinders—

$$\left(\frac{L}{l} = 1{\cdot}8 \text{ to } 2\right)$$

If these conditions cannot be secured in practice, as sometimes happens, because the advantages of some other arrangement cannot be

sacrificed merely to secure more effective balancing, the outer connecting rods must be artificially lightened by means of counterweights, and the connecting rods of the middle cylinders artificially weighted, by the use of heavy pistons. Besides this, in engines having a ratio $\frac{L}{l} < \frac{4}{3}$ the angles of the two outer cranks are apt to be rather small. This has an injurious effect on the manœuvring of the engines, and the regularity of the turning moment (see pages 125 and 63). But if the heavy rods be in the middle and $\frac{L}{l}$ = about 2, the best crank angles will be obtained, the smallest being about 65° to 70°, and perfect balance can be secured without adding extra weights (see following example). In triple expan-

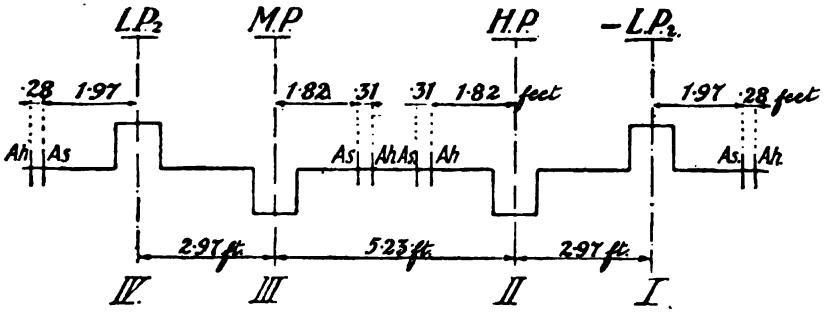


Fig. 56.

sion engines good balancing is most easily obtained by having two L.P. cylinders, each doing purposely less work, and having lighter connecting rods than the H.P. and M.P. cylinders, outside which they are placed (Yarrow-Schlick-Tweedy system). An engine with four cranks, in which the forward pair of cranks is at right angles to the after pair, and the cranks of each pair are at an angle of 180° to each other, gives an almost perfect balance, if the distance between the two pairs of cranks is relatively large in proportion to the distance between the individual cranks constituting each pair, and when the masses of the forward pair are equal to the masses of the after pair. In engines running at high speeds, the balance is considerably affected by the relatively heavy parts of the valve gear, and their long stroke. A

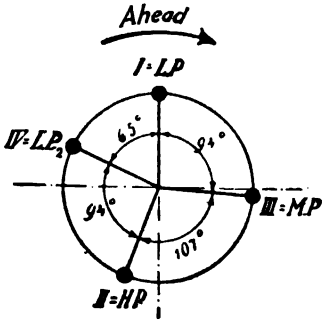


Fig. 57.

better balance is obtained in such engines, by placing the valve gear outside the end cranks, and carefully proportioning the steam and exhaust ports of the slide valve.

§ 44. **First Example: Calculation of the Balance of the Moving Parts in an Existing Engine.**—This engine was built on the Yarrow-Schlick-Tweedy system, the balancing of the moving parts being specially considered. The distance between the two middle cylinders is relatively great. The connecting rods of the L.P. cylinders are specially light, in view of the less amount of work to be done in these cylinders. Fig. 56 shows the arrangement of the engine, Fig. 57 the positions of the cranks, Fig. 58 positions of the eccentrics.

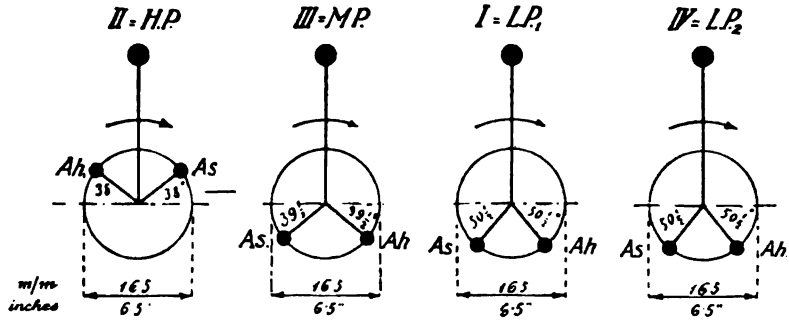


Fig. 58.

Weights of the Connecting Rods.

Details.	IV = No. 2 L.P. Cylinder.	III = M.P. Cylinder.	II = H.P. Cylinder.	I = No. 1 L.P. Cylinder.
Reciprocating Parts—	lb.	lb.	lb.	lb.
Piston	588·47	606·10	312·96	588·47
Piston rod and crosshead	427·57	606·10	606·10	427·57
Connecting rod (top end)	184·47	295·33	295·33	184·47
Pump rods	−50·25	...	...	...
Total	1150·26	1507·53	1214·39	1200·51
Rotating Parts—				
Connecting rod (bottom end)	319·58	418·76	418·76	319·58
Crank pin and cheeks	780·21	1055·71	1055·71	780·21
Total	1099·79	1474·47	1474·47	1099·79
Total weight of the reciprocating and rotating parts	2250·05	2982·00	2688·86	2300·30

Weights of the Valve Gear.

Details.	IV = No. 2 L.P. Cylinder.		III = M.P. Cylinder.		II = H.P. Cylinder.		I = No. 1 L.P. Cylinder.	
	Ahead.	Astern.	Ahead.	Astern.	Ahead.	Astern.	Ahead.	Astern.
	lb.	lb.	lb.	lb.	lb.	lb.	lb.	lb.
Reciprocating Parts—								
Slide valve, valve rod, and link block	498·11	...	537·78	...	277·51	...	498·11	...
Half of reversing link - - -	35·04	35·04	54·22	54·22	54·22	54·22	35·04	35·04
2 × ½ reversing rod - - - -	24·02	...	34·16	...	34·16	...	24·02	...
Eccentric rod (top end) - - -	55·10	55·10	81·32	81·32	81·32	81·32	55·10	55·10
Total -	612·27	90·14	707·48	135·54	447·41	135·54	612·27	90·14
Rotating Parts—								
Eccentric strap - - - - - -	147·68	140·39	198·80	191·75	198·80	191·75	147·68	140·39
Eccentric rod (bottom end) - -	25·12	25·12	38·13	38·13	38·13	38·13	25·12	25·12
Eccentric - - - - - -	137·52	127·40	184·69	172·57	184·69	172·57	137·52	127·40
Total -	310·32	292·91	421·62	402·45	421·62	402·45	310·32	292·91
Total weight of the rotating and reciprocating parts - - - - - -	922·59	383·05	1129·10	537·99	869·03	537·99	922·59	383·05

HORIZONTAL BALANCE—MOVEMENT OF VALVE GEAR ABOUT PLANE I.

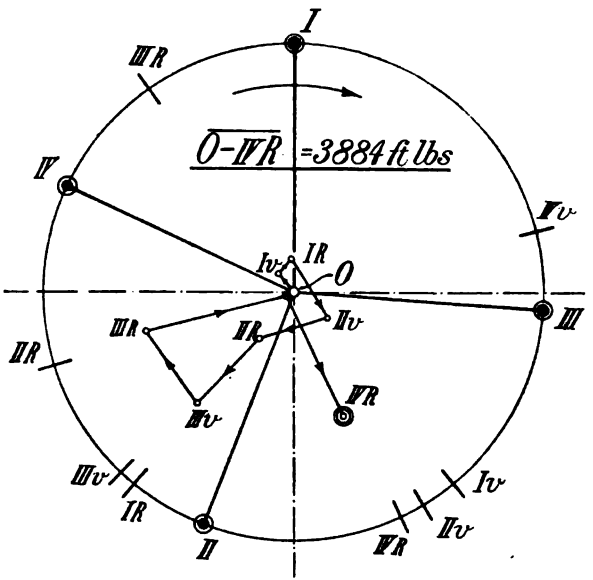


Fig. 60.

VERTICAL BALANCE—MOVEMENT OF VALVE GEAR ABOUT PLANE I.

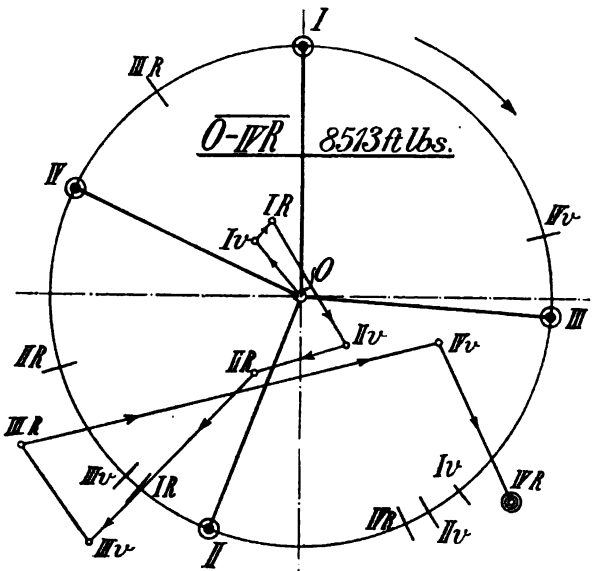


Fig. 59.

VERTICAL BALANCE—MOVEMENT OF VALVE GEAR ABOUT PLANE IV.

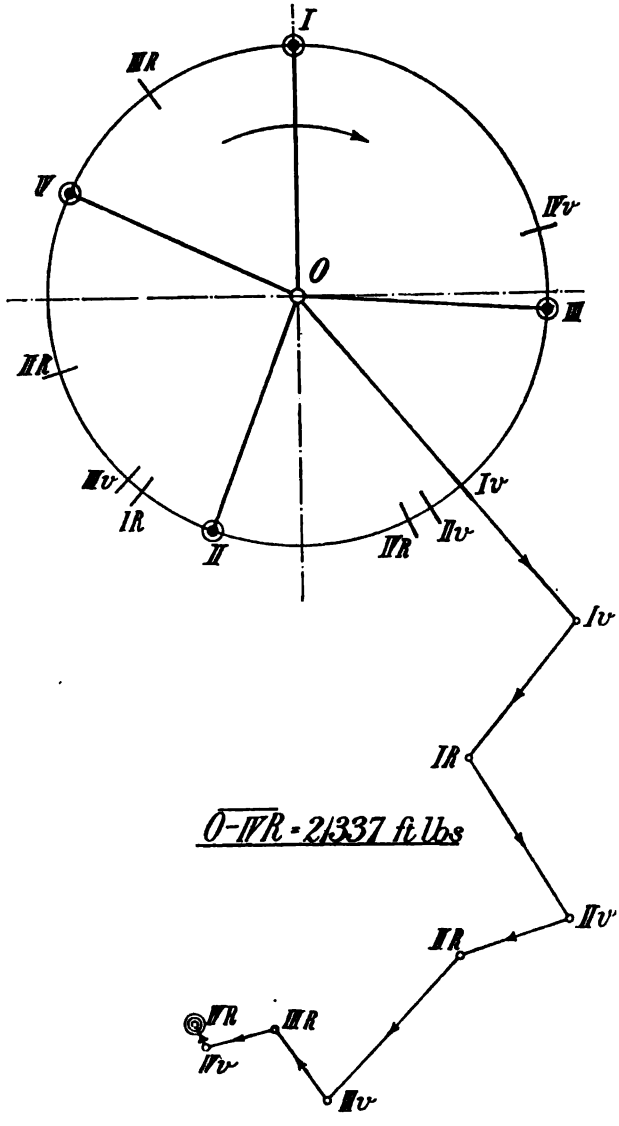


Fig. 61.

HORIZONTAL BALANCE—MOVEMENT OF VALVE GEAR ABOUT PLANE IV.

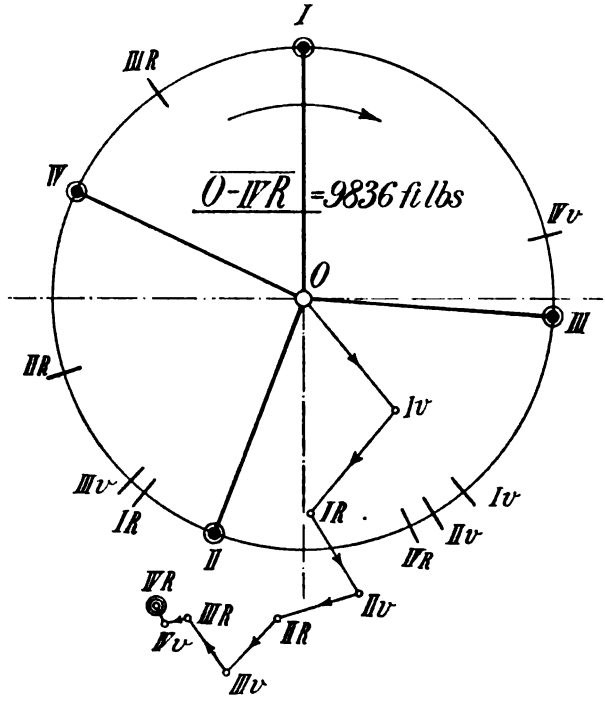


Fig. 62.

In the following calculations the actual weights are used, instead of the masses of the moving parts, because the factor $\frac{w^2}{r} \cdot \frac{1}{g}$ is common to all the latter (see page 87). The weights of the moving parts of the valve gear are first reduced to the circle described by the eccentric, and the resultant reduced to the circle described by the crank.

We will work out first the resultant moment of the moving parts of the valve gear, as follows :—

1. For the vertical balance about plane I.
2. For the vertical balance about plane IV.
3. For the horizontal balance about plane I.
4. For the horizontal balance about plane IV.

Moments of the Moving Parts of the Valve Gear about Plane I.

	Vertical Balance.		Horizontal Balance.	
	lb. ft.	ft. lb.	lb. ft.	ft. lb.
I Ahead	922·59 × (−2·25) =	−2075·8	310·32 × (−2·25) =	−697·98
Astern	383·05 × (−1·97) =	− 759·4	292·91 × (−1·97) =	− 578·64
II Ahead	869·03 × 4·8 =	4173·4	421·62 × 4·8 =	2025·24
Astern	537·99 × 5·11 =	2755·7	402·45 × 5·11 =	2061·40
III Ahead	1129·10 × 6·08 =	6872·3	421·62 × 6·08 =	2567·71
Astern	537·99 × 6·39 =	3442·9	402·45 × 6·39 =	2574·94
IV Ahead	922·59 × 13·42 =	12411·8	310·32 × 13·42 =	4173·44
Astern	383·05 × 13·16 =	5048·6	292·91 × 13·16 =	3862·42

Moments of the Moving Parts of the Valve Gear about Plane IV.

	Vertical Balance.		Horizontal Balance.	
	lb. ft.	ft. lb.	lb. ft.	ft. lb.
I Ahead	922·59 × 13·42 =	12411·8	310·32 × 13·42 =	4173·44
Astern	383·05 × 13·16 =	5048·6	292·91 × 13·16 =	3862·42
II Ahead	869·03 × 6·38 =	5547·7	421·62 × 6·38 =	2690·67
Astern	537·99 × 6·07 =	3269·3	402·65 × 6·07 =	2444·75
III Ahead	1129·10 × 5·10 =	5750·2	421·62 × 5·10 =	2148·20
Astern	537·99 × 4·79 =	2574·9	402·65 × 4·79 =	1923·97
IV Ahead	922·59 × (−2·25) =	−2075·8	310·32 × (−2·25) =	697·98
Astern	383·05 × (−1·97) =	− 759·4	292·91 × (−1·97) =	578·64

These moments are combined in the polygons shown in Figs. 59 to 62, regard being had to the positions of the eccentrics. The lines $\overline{\text{O IV R}}$ are the resultant moments of each polygon, *i.e.*, the moment $\overline{\text{O IV R}}$ represents the resultant action of the different moments of which it is composed.

We are now able to plot the polygon of moments of the cranks and valve gear about planes I and IV, to arrive at the horizontal and vertical balance. If the lines thus produced form a closed figure, the balance of the parts is complete.

Moments of the Connecting Rods about Plane I.

Crank	Vertical Balance.	Horizontal Balance.
	lb. ft. ft. lb.	lb. ft. ft. lb.
II	2688·8 × 2·97 = 8021·39	1474·4 × 2·97 = 4377·6
III	2982·0 × 8·20 = 24462·0	1474·4 × 8·20 = 12115·2
IV	2250·2 × 11·17 = 25170·8	1099·8 × 11·17 = 12325·0

Moments of the Connecting Rods about Plane IV.

Crank	Vertical Balance.	Horizontal Balance.
	lb. ft. ft. lb.	lb. ft. ft. lb.
I	2300·9 × 11·17 = 25749·4	1099·7 × 11·17 = 12325·0
II	2688·8 × 8·20 = 22075·1	1474·4 × 8·20 = 12115·2
III	2982·0 × 2·97 = 8889·3	1474·4 × 2·97 = 4397·6

1. *Vertical Balance: Moments reduced to Plane* I.—Fig. 63 shows the polygon combining the moments of the connecting rod for vertical reciprocation, as reduced to plane I, with the corresponding sum of the moments $\overline{\text{O IV R}}$ of the valve gear. $\overline{\text{O IV R}}$ = 8,509 ft. lb. (reduced to the radius of the eccentric) must be reduced by the ratio of the radius of the eccentric (3·24 inches) to that of the crank (14·99 inches) in order to combine it with the moments of the crank. The moment of the valve gear, thus reduced, is $\overline{\text{IV O}}$ = 1,844 ft. lb. Fig. 63 shows that the polygon of moments O II III IV O forms a closed figure, and therefore the moment of the vertical balance of the moving parts, reduced to plane I, disappears.

2. *Vertical Balance: Moments reduced to Plane* IV (Fig. 64).—$\overline{\text{O IV R}}$ = 21,337 ft. lb., moment of the valve gear, reduced to the radius of the eccentrics. $\overline{\text{III O}}$ = 4,629 ft. lb., is the same, reduced to the circle described by the crank. As the polygon O I II III O also forms a closed figure, the vertical balance is therefore complete.

3. *Horizontal Balance: Moments reduced to Plane* I (Fig. 65).—

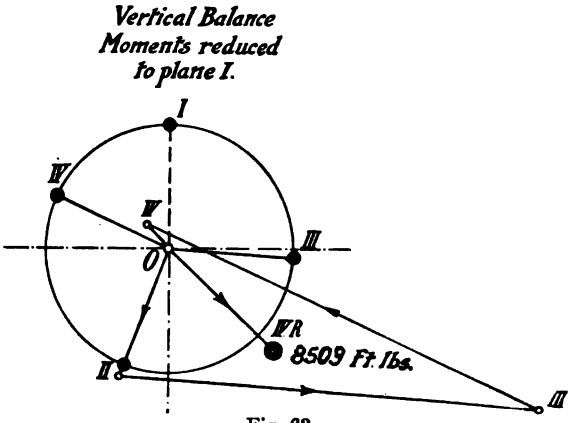


Fig. 63.

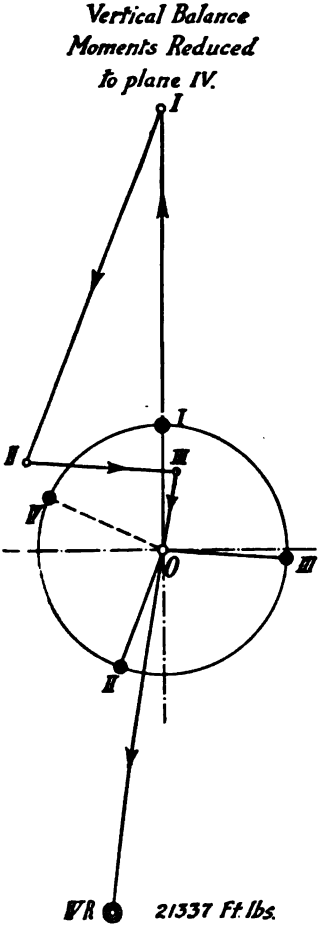


Fig. 64.

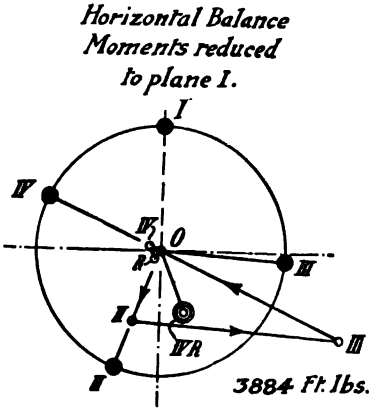


Fig. 65.

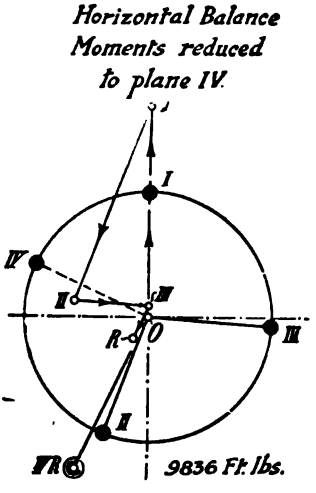


Fig. 66.

o IV R = 3,884 ft. lb., moment of the valve gear, reduced to the radius of the eccentric. $\overline{\text{IV R}}$ = 839 ft. lb., is the same, reduced to the radius of the crank. Here the polygon of moments o II III IV R is not closed; there is thus an incomplete balance of the moving parts = $\overline{\text{OR}}$ = 578 ft. lb. (reduced to the crank circle).

4. *Horizontal Balance: Moments reduced to Plane* IV (Fig. 66).—$\overline{\text{o IV R}}$ = 9,836 ft. lb., moment of the valve gear, reduced to the circle described by the eccentric. $\overline{\text{III R}}$ = 2,133 ft. lb., is the same, reduced to the crank circle. Here also the polygon o I II III R is not a closed figure, and the balance is incomplete by $\overline{\text{OR}}$ = 1,519 ft. lb. The engine has thus perfect vertical and almost perfect horizontal balance. The small discrepancies in the latter may be neglected. The balance is obtained without counterweights, and without weighting the piston.

§ 45. **Second Example: Calculation of the Balance of the Moving Parts in the Engine of a Fast Mail Steamer.**—The arrangement of the cylinders and valve gear are given, and for constructive reasons must be adhered to, although it does not give the best balance. It is usually stipulated that the different parts of the crank shaft shall be interchangeable. The holes for the coupling bolts must therefore be bored in such a way that the angles of the crank are multiples of the angles of the bolt holes. For the arrangement of the cylinders see Fig. 67.

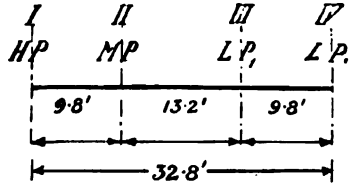


Fig. 67.

Weights and Moments of the Crank Rods, as determined by Constructive Conditions.

		H.P.	M.P.	No. 1 L.P.	No 2 L.P.
Weights for calculating the vertical balance -	tons	18·8	20·7	20·7	20·7
Weights for calculating the horizontal balance	,,	11·0	11·0	11·0	11·0
Moments for calculating the vertical balance about plane I - - - - - - - -	ft. tons	...	203	455	...
Moments for calculating the vertical balance about plane IV - - - - - - - -	,,	...	455	203	...
Moments for calculating the horizontal balance about plane I - - - - - - - -	,,	...	108·4	253	...
Moments for calculating the horizontal balance about plane IV - - - - - - - -	,,	...	253	108·4	...

A cursory examination shows that in this case the crank positions given in Fig. 68 are the best, and they also satisfy the condition for the interchangeability of the coupling bolts, if fourteen bolts are put into each flange. The positions of the crank being thus fixed beforehand, the resultant moment of the valve gear can be determined. This is done in exactly the same way as in Example I., and only the results obtained are given here; the moments of the valve gear about planes I and IV for the vertical balance s_1^v and s_4^v are drawn out in Fig. 69, and for the horizontal balance s_1^h and s_4^h of the valve gear in Fig. 70. As already observed, this arrangement of the cylinders is not favourable to good balancing, so it is necessary to increase the weight of the centre pistons, and to fit counterweights on the outer cranks. The amount of additional weight required is deduced from Figs. 69 and 70.

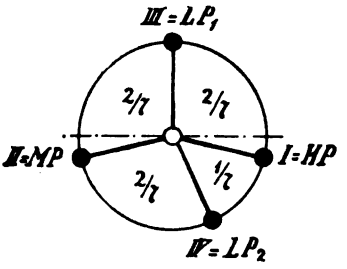


Fig. 68.

1. *Horizontal Balance: Determination of the Counterweights* (Fig. 70).—The moment of crank I^h must annul those of cranks II^h and III^h about plane IV. $o\,I_1^h$ is thus = the resultant of the moments $o\,II_1^h$ and $o\,III_1^h$. But crank I^h must further annul the moments of the valve gear about plane IV. Therefore to the resultant $o\,I_1^h$ we must add the resulting moments of the valve gear for the horizontal balance in plane IV, namely, $I_1^h\,I_2^h = oS_1^h$. If balance were complete, $o\,I_2^h = 226$ foot tons would be the moment of the masses of crank I actually rotating about plane IV. For constructive reasons (compare the Table of Weights) the moment of crank I^h about plane IV has to be $o\,I^h = 361{\cdot}4$ foot tons. Thus there is a discrepancy, $I_2^h\,I^h$, in the balance of the parts. This may be eliminated by a counterweight on crank I, the dimensions and direction of which are $I^h\,I_2^h$. The moment of this counterweight about plane IV is $oG_1 = I^h\,I_2^h = 132{\cdot}3$ foot tons, and the weight of the counterweight is therefore $\frac{132{\cdot}3}{32{\cdot}8} = 4{\cdot}0$ tons. The dimensions and direction of the counterweight on crank IV are found in the same way.

Moment $G_4 = IV^h\,IV_2^h = 83{\cdot}8$ foot tons, weight $G_4 = 2{\cdot}56$ tons.

The addition of these weights makes the horizontal balance of the parts complete.

2. *Vertical Balance: Determination of the Weight required on the Piston* (Fig. 69).—$o\,I_1^v$ = the resultant of the vertical moments of cranks II and III about plane IV ($o\,II_1^v$ and $o\,III_1^v$); $o\,IV_1^v$ also = the resultant

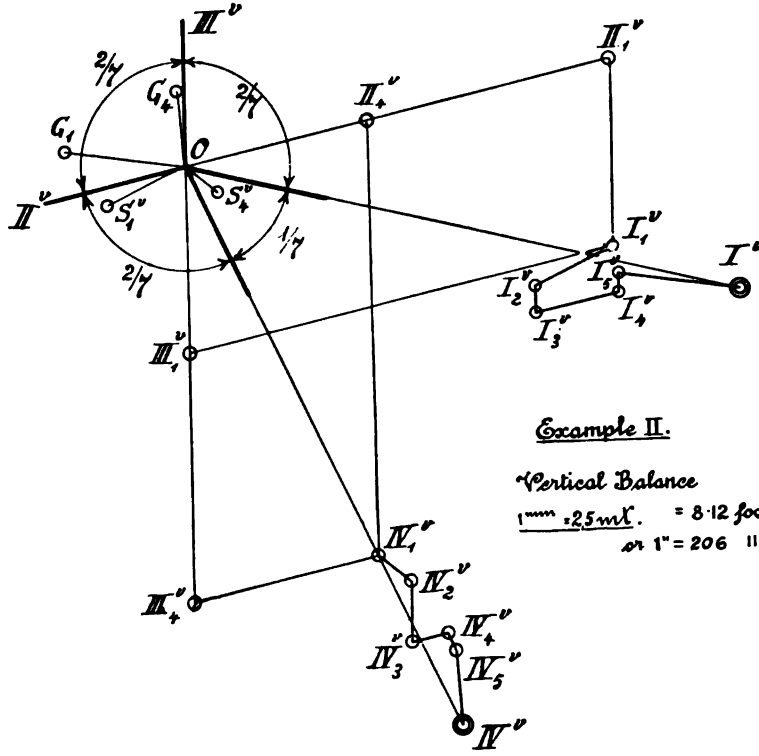

III v
II v
II 1 v
G 4
G 1
O
2/7
2/7
2/7
2/7
1/7
II 4 v
S 1 v
S 4 v
I 1 v
I v
I 5 v
I 2 v
I 4 v
I 3 v
III 1 v
Example II.
Vertical Balance
1mm = 2,5 m/t. = 8·12 foot tons
or 1" = 206 " "
IV 1 v
III 4 v
IV 2 v
IV 4 v
IV 3 v
IV 5 v
IV v

Fig. 69.

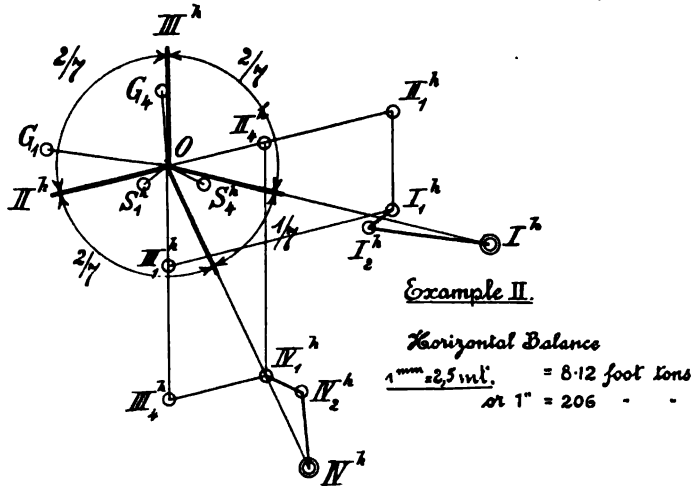

III h
2/7
2/7
G 4
II 1 h
G 1
II 4 h
O
II h
S 1 h
S 4 h
I 1 h
I h
1/7
2/7
II 1 h
I 2 h
Example II.
Horizontal Balance
1mm = 2,5 m/t. = 8·12 foot tons
or 1" = 206 " "
IV 1 h
III 4 h
IV 2 h
IV h

Fig. 70.

of the vertical moments of cranks II and III about plane I ($o\,II_4^v$ and $o\,III_4^v$). The moments of the valve gear about plane IV and plane I respectively must compensate for cranks I and IV. These moments are os_1^v and os_4^v, and are added to I_1^v and IV_1^v, so that $os_1^v = I_1^v\ I_2^v$ and $os_4^v = IV_1^v\ IV_2^v$. The moments of the cranks $o\,I^v$ and $o\,IV^v$ are determined by constructive conditions; the counterweights required for horizontal balance must be deduced from them. If they are added to I^v and IV^v respectively, we get the points I_5^v and IV_5^v. If this gave the required vertical balance, $o\,I_2^v$ and $o\,I_5^v$, and $o\,IV_2^v$ and $o\,IV_5^v$ would be respectively equal to each other. This is not the case; and since the counterweights cannot be altered, because that would affect the horizontal balance, the two centre pistons must be weighted. For piston III the weight must be such that its moments about plane $IV = I_2^v\ I_3^v$, and about plane $I = IV_2^v\ IV_3^v$; for piston II the moments of the weight about planes IV and I must be $= I_3^v\ I_4^v$ and IV_3^v and IV_4^v respectively.

With the pistons thus loaded there are still discrepancies in the vertical balance, but they are not important; they consist of the resulting moments $I_4^v\ I_5^v$ and IV_4^v and IV_5^v, which cannot be eliminated by weighting the pistons, because if the load on piston III is increased the moment $IV_4^v\ IV_5^v$ is reduced, but the moment $I_4^v\ I_5^v$ is augmented. This very small discrepancy might be completely avoided if the angles of the cranks and the counterweights could be slightly varied.

The weight on piston III is—

$$\frac{I_2^v\ I_3^v}{9{\cdot}8} = \frac{IV_2^v\ IV_3^v}{22{\cdot}9} = \frac{29{\cdot}0}{9{\cdot}8} = \frac{67{\cdot}7}{22{\cdot}9} = 2{\cdot}95 \text{ tons.}$$

The weight on piston II is—

$$\frac{I_3^v\ I_4^v}{22{\cdot}9} = \frac{IV_3^v\ IV_4^v}{9{\cdot}8} = \frac{90{\cdot}3}{22{\cdot}9} = \frac{38{\cdot}7}{9{\cdot}8} = 3{\cdot}94 \text{ tons.}$$

§ 46. **Effect of the Length of the Connecting Rod on the Balance of the Moving Parts.**—The above deductions for the vertical balance are valid for "connecting rods of infinite length." In practice, for connecting rods of a given length, the vertical balance of the moving parts does not follow the law $\frac{M}{g}\frac{W^2}{r}\cos\omega$, but approximates to the law $\frac{M}{g}\frac{W^2}{r}(\cos\omega + \lambda\cos 2\omega)$. See page 62.

If in the sum of the moments for vertical balance (Equations 1 and 2, § 39) we substitute $\cos\omega + \lambda\cos 2\omega$, $\cos(\alpha+\omega) + \lambda\cos 2(\alpha+\omega)$, &c., for

$\cos\omega$, $\cos(\alpha+\omega)$, &c., these totals will not be $=0$,* even in engines with complete balance, and connecting rods of "infinite length."

These totals may then be written thus: (1) Moments about plane I—

$$M_2a\cos(\omega+\alpha+\beta+\gamma)+M_3(a+b)\cos\omega+M_4(a+b+c)\cos(\omega+\alpha+\beta)+$$
$$M_2a\,.\,\lambda\cos 2(\omega+\alpha+\beta+\gamma)+M_3(a+b)\lambda\cos 2\omega+$$
$$M_4(a+b+c)\lambda\cos 2(\omega+\alpha+\beta)$$

and (2) moments about plane IV—

$$M_3c\cos\omega+M_2(b+c)\cos(\omega+\alpha+\beta+\gamma)+M_1(a+b+c)\cos(\omega+\alpha)+$$
$$M_3c\lambda\cos 2\omega+M_2(b+c)\lambda\cos 2(\omega+\alpha+\beta+\gamma)+$$
$$M_1(a+b+c)\lambda\cos 2(\omega+\alpha).$$

In every engine where the vertical balance of the moving parts is complete, and with connecting rods of "infinite length," the three first terms of each of these equations disappear, and the three last terms may be combined into one resultant moment. These moments attain a maximum and a minimum twice in every revolution, and therefore they can only produce vibrations, the number of which is equal to twice the number of revolutions of the engine.

No vibrations to any considerable extent, produced by these moments, have been detected in vessels, although the existence of the vibrations has often been proved by means of the "Pallograph." (See Part VI.) They are caused by the length of the connecting rod, and to reduce them within small limits a special modification of the balanced parts has been introduced by Schlick; but want of space forbids a description of it.

§ 47. **Critical Number of Revolutions and Effect of the Position of the Engine on the Vibrations of the Hull.**†— The ship's hull may be considered as an elastic beam or girder supported throughout its whole length upon an elastic medium. Such a beam will "vibrate" to its greatest extent, if it does so at all, in the manner shown at Fig. 71, the curves of vibration forming nodes at the two points A and B. The engine produces the greatest vibratory effect when the number of its revolutions (and therefore of vibratory impulses) coincides with the natural vibrations of the ship. The number of revolutions which

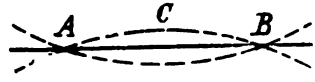


Fig. 71.

* In four-crank engines only one of these totals can be $=0$ with connecting rods of a given length.

† See Schlick, *Zeitschrift des Vereines Deutscher Ingénieure*, 1894, vol. ii., p. 1091.

produces the maximum vibratory effect of the first order is called the "*critical*" *number of revolutions*, and is approximately *—

$$n = k\sqrt{\frac{T}{DL^3}}$$

In this formula T signifies the moment of inertia of the mid-ship section, to get which the cross-sections must be given in square feet, and the arm in feet; D the displacement, in tons; L the length of the ship on the water line, in feet; and k is a coefficient having the following values—

$k = 34050$ for torpedo-boat destroyers;
$= 31200$ for large, fast steamers with fine lines; and
$= 27800$ for "full" cargo vessels.

Position of the engine in the ship. An engine with free moment of the moving parts *would tend to accentuate vibrations* of the nature shown in Fig. 71, when it is at the points A or B, and *would not materially influence them* when it is in the centre at C.

An engine with free resultant of the balance of the moving parts *would tend to increase the vibrations* shown in Fig. 71, when it is in the centre at C, and *has no similar tendency* when it is at the points A or B.

* See Schlick, *Trans. of the Inst. of Naval Architects*, 1894, p. 350.

SECTION IV.

ARRANGEMENT OF MAIN ENGINES.

§ 48. **Arrangement of the Cylinders and Cranks.**—1. *Twin-cylinder engines* (sometimes used instead of compound engines for small and very light boats, such as pinnaces) have two cylinders, either cast in one or else solidly bolted together, and two cranks at an angle of 90°.

2. *Compound engines* (for small freight steamers and passenger steamers for river traffic, &c., with engines from 30 to 500 I.H.P., or even up to 1,500 I.H.P.). These are now only made with one H.P. and one L.P. cylinder, and with two cranks at an angle of 90°. Formerly in compound engines with more than two cylinders several different arrangements were adopted, and they were often used for higher powers with one H.P. and two L.P. cylinders, and three cranks at an angle of 120°. (Fast steamers "Ems" and "Elbe" with engines of 6,000 I.H.P.)

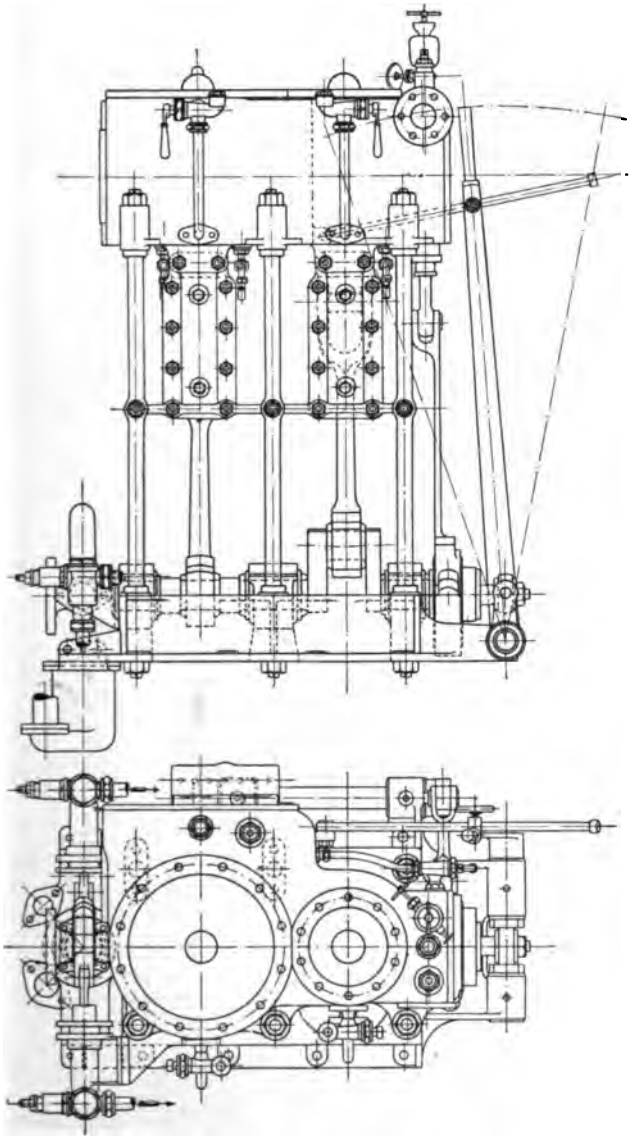
Figs. 72, 73, 74, 75 give four views of the engine of a ship's pinnace. I.H.P. about 50; number of revolutions per minute = 300. Diameter of H.P. cylinder, 160 mm. (6·29 inches); L.P. cylinder, 300 mm. (11·81 inches); stroke, 200 mm. (7·87 inches); boiler pressure, 142 lb. per square inch. The reversing gear is worked by a hand lever, altering a single adjustable eccentric for both cylinders. Air, bilge, and feed pumps are also driven by one eccentric.

Figs. 76, 77, and 78 show a rather heavier *engine of a small freight steamer.* I.H.P. about 200; revs. = 130. Diameter of H.P. cylinder, 380 mm. (14·96 inches); L.P. cylinder, 680 mm. (26·77 inches); stroke, 500 mm. (19·68 inches); boiler pressure, 114 lb. per square inch. Stephenson's link motion, the reversing link being moved by a screwed spindle and hand wheel. Air, bilge, and feed pumps are driven direct from the L.P. crosshead by means of a lever.

3. *Triple-expansion engines* are now almost universally used for all kinds of marine engines developing more than 300 I.H.P. The types most generally employed are—

(*a.*) *Three-crank engines, with one* H.P., *one* M.P., *and one* L.P. *cylinder, the cranks being at* 120°. This arrangement is employed on all kinds of cargo steamers, passenger boats, warships, and torpedo-boats, but it cannot be used for very high powers, because the dimen-

sions of the L.P. cylinder become too large and unwieldy. The risks and difficulties of machining and fitting such heavy castings are so great, that



Figs. 72 and 73.

the diameter of the L.P. cylinder is seldom more than from 8 feet 10 inches to 9 feet 2 inches.

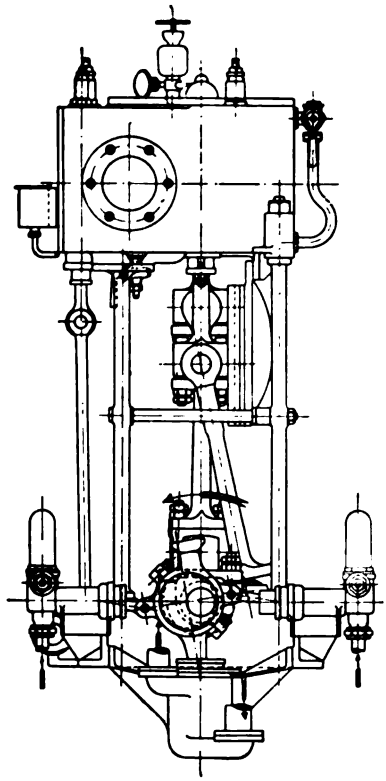


Fig. 74.

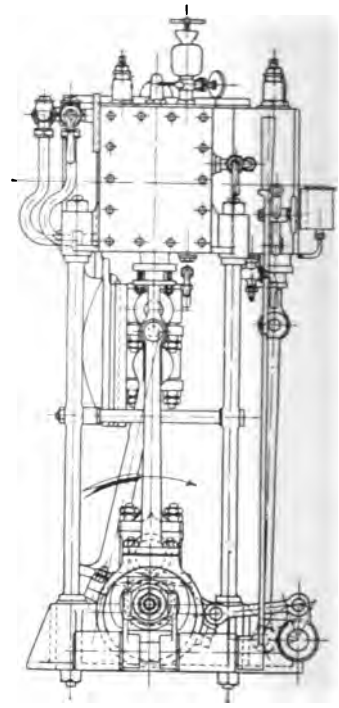


Fig. 75.

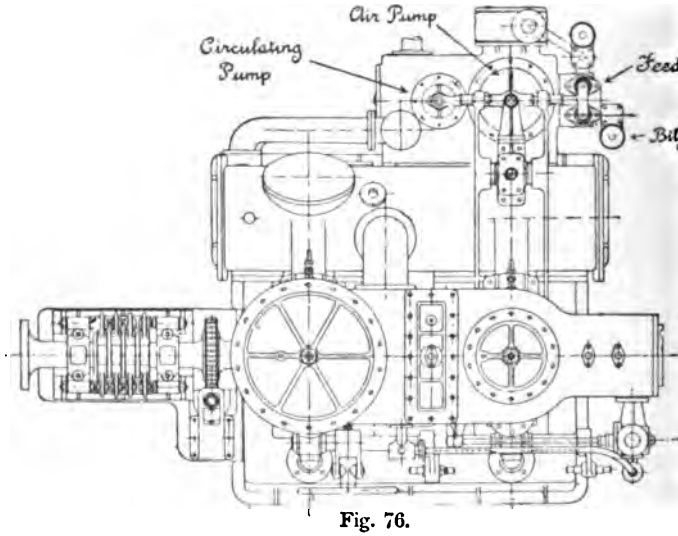


Fig. 76.

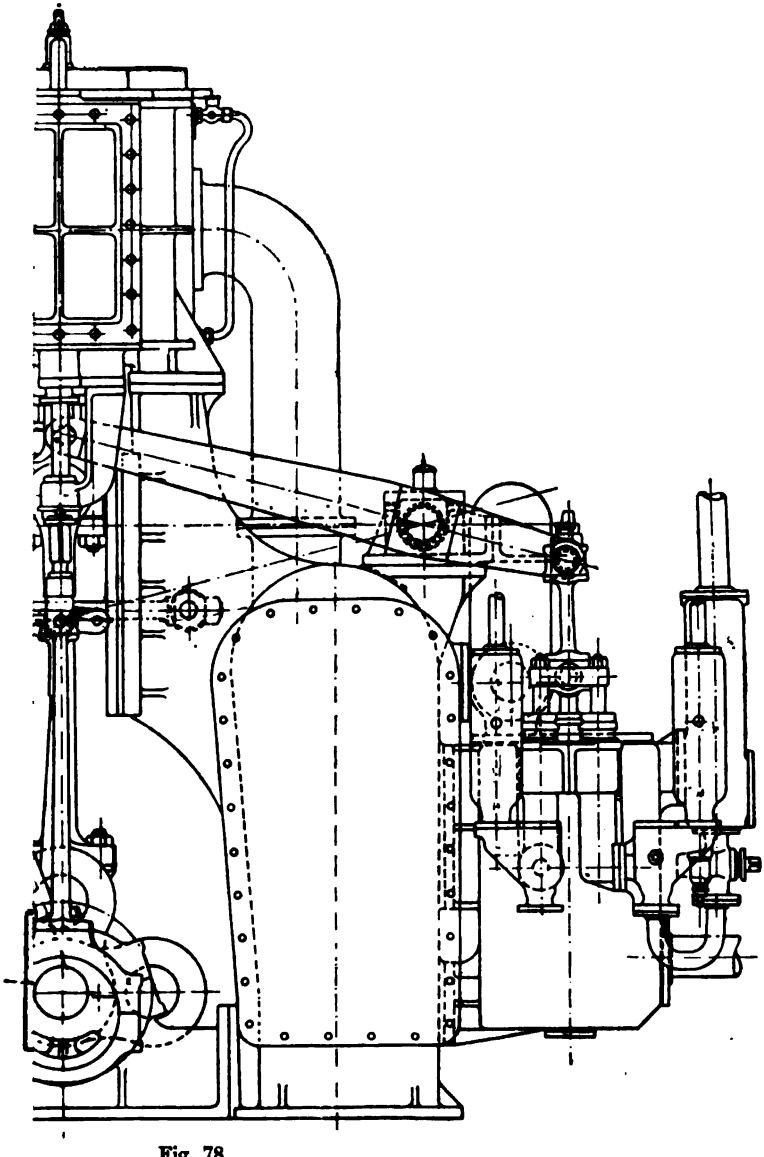


Fig. 78.

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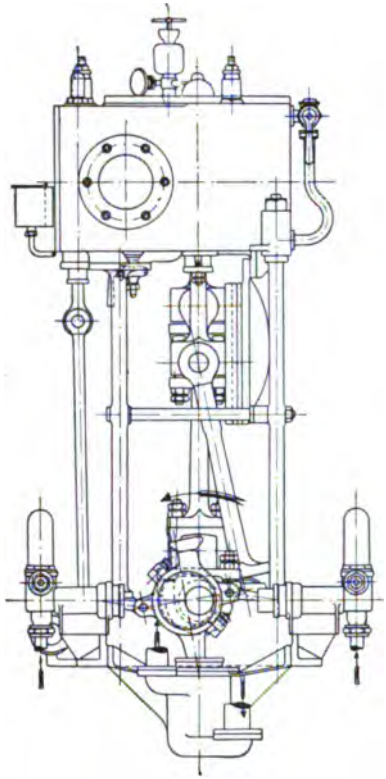

Fig. 74.

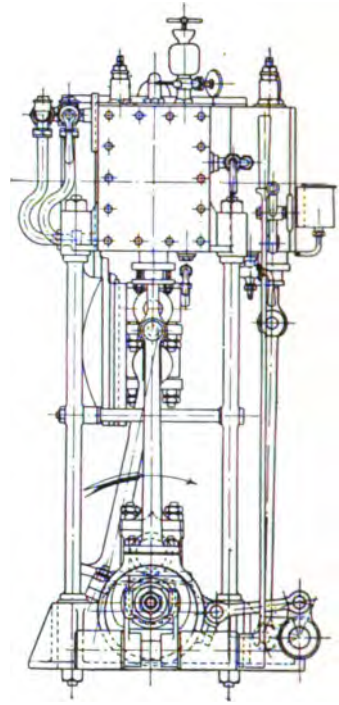

Fig. 75.

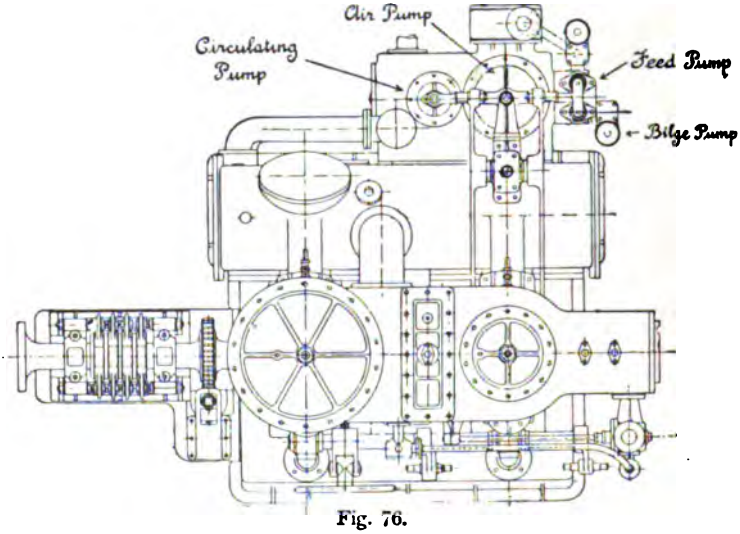

Fig. 76.

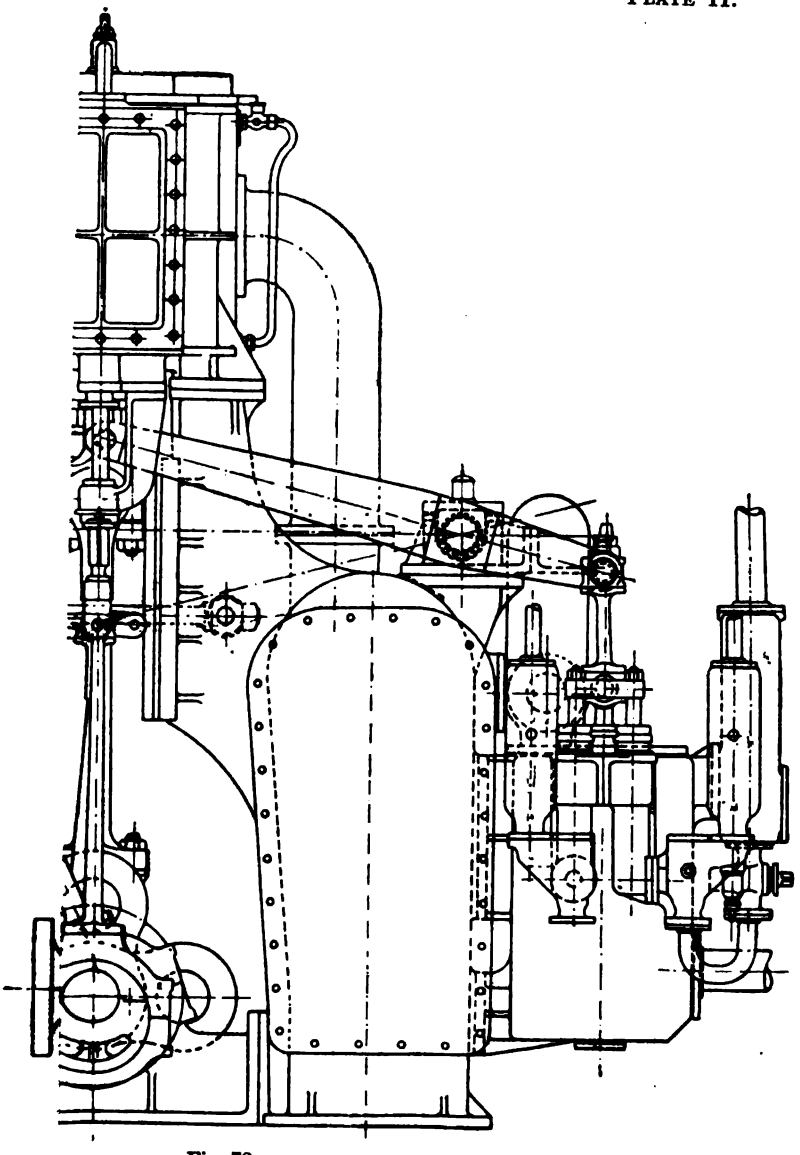

Fig. 78.

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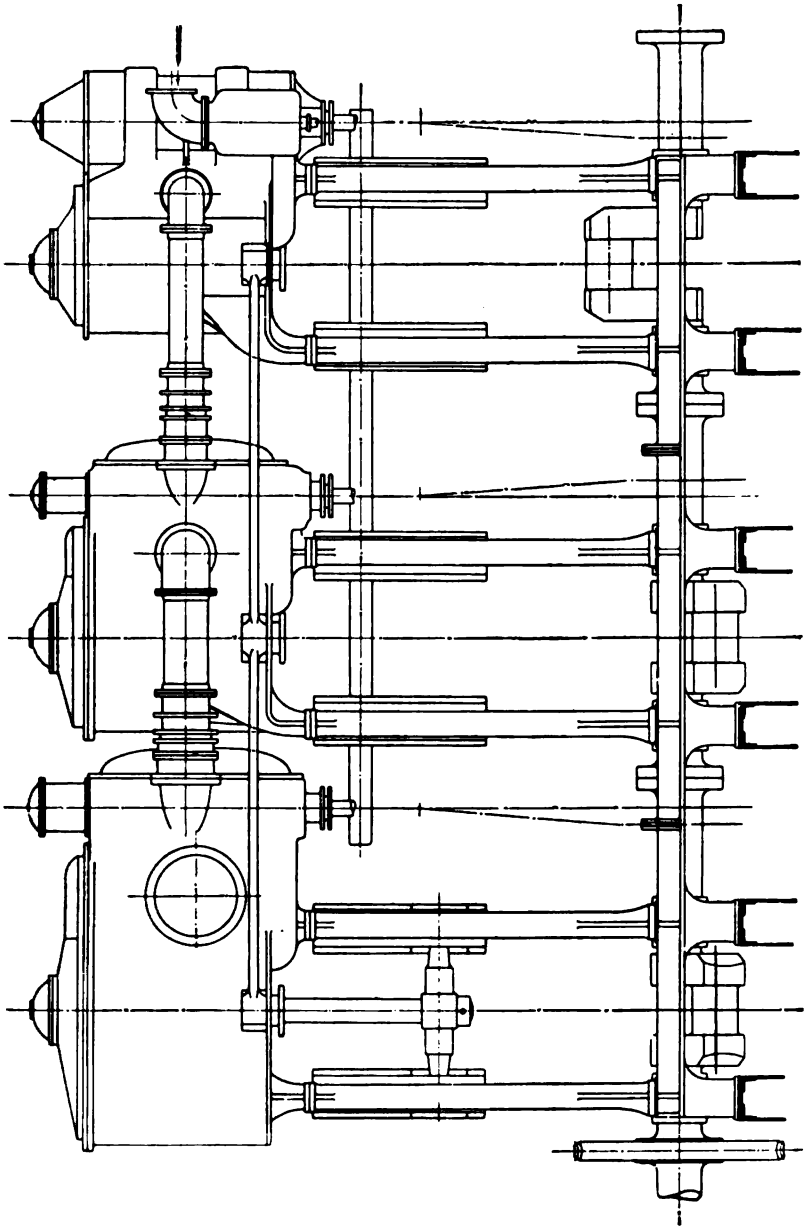

Fig. 79.

Figs. 82 to 84 show *the engine of a steam trawler.* I.H.P. about 400. Cylinder diameters, 305, 483, 819 mm. (12 inches, 19 inches, 32¼ inches) respectively; stroke, 610 mm. (24 inches); revs. = 140; boiler pressure, 164 lb. per square inch. Stephenson's link motion with crossed eccentric rods. Air, circulating, bilge, and feed pumps are worked by a lever from the M.P. crosshead.

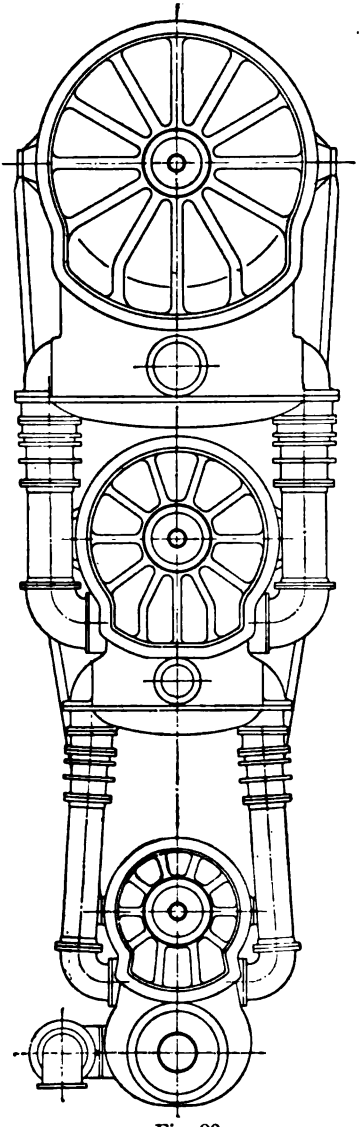

Fig. 80.

Figs. 79 to 81 show the *engines of a battleship* with twin screws. I.H.P. = 2 × 6,000; revs. = 100. Diameter of cylinders, 1,016, 1,498, 2,235 mm. (40 inches, 59 inches, 88 inches); stroke, 1,295 mm. (51 inches); boiler pressure, 156 lb. per square inch. Stephenson's link motion; condenser at the side of the engine. Each cylinder is supported by four cast-steel columns.

Figs. 85 to 87. *Engines of the twin-screw mail steamer* "Fürst Bismarck." I.H.P. = 2 × 8,200; revs. = 85. Diameters of cylinders, 1,100, 1,700, 2,700 mm. (3 feet 7¼ inches, 5 feet 7 inches, 8 feet 10 inches); stroke, 1,600 mm. (5 feet 3 inches); boiler pressure, 156 lb. per square inch. Stephenson's link motion. Each cylinder is supported on two cast-iron **A** frames, one of which rests on the cast-iron condenser.

Fig. 88, Plate III. *Engines of the Imperial yacht* "Hohenzollern." I.H.P. = 2 × 4,500. The reversing gear is on the Klug system; condenser at the side of the engine. Each cylinder is supported on two cast-steel columns.

(*b.*) *Three-crank engines with five cylinders*, namely, *two* H.P., *one* M.P., *and two* L.P. *cylinders.* Angle of the cranks, 120°. The centre crank

is driven by the M.P. cylinder; each of the two outer cranks is driven by one of the L.P. cylinders, upon the cover of one of which an H.P. cylinder is placed tandem. This arrangement has been carried out in several large mail steamers, such as the "Campania" and "Lucania," which have two engines each indicating 14,000 H.P.; also on the "Spree" and the "Havel," each of which has a single engine indicating 12,500 H.P.

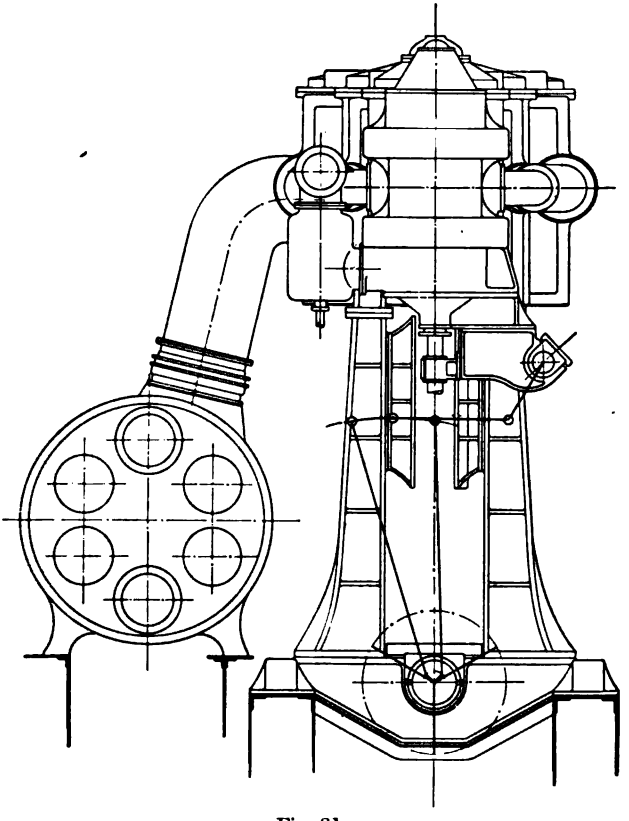

Fig. 81.

(*c.*) *Four-crank engines with one* H.P., *one* M.P., *and two* L.P. *cylinders.* This arrangement gives long, narrow engines; the cranks are usually set at an angle of 90°, or balanced in accordance with the Schlick system. The advantages of this arrangement are many; the division of the L.P. cylinder, necessary in large engines, is obtained without having to place one cylinder above another; and in small war ships, where the engines must not project above the armoured deck, large cylinders can be avoided, and the two engines put side by side. When the Schlick

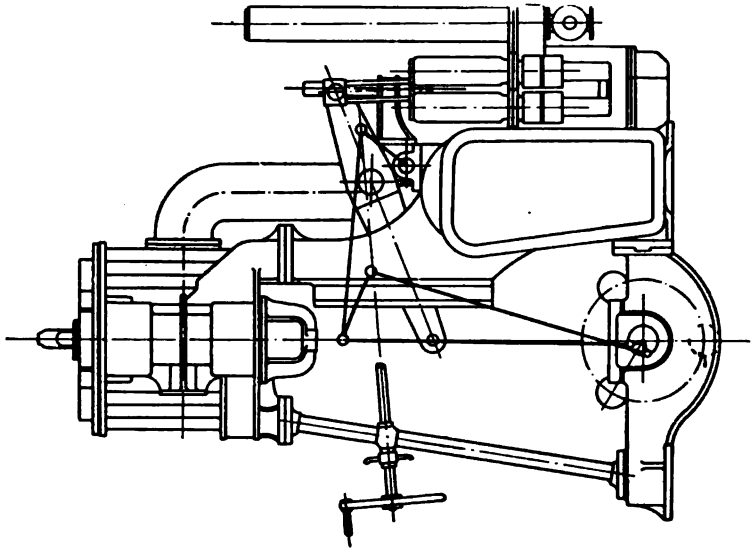

Fig. 83.

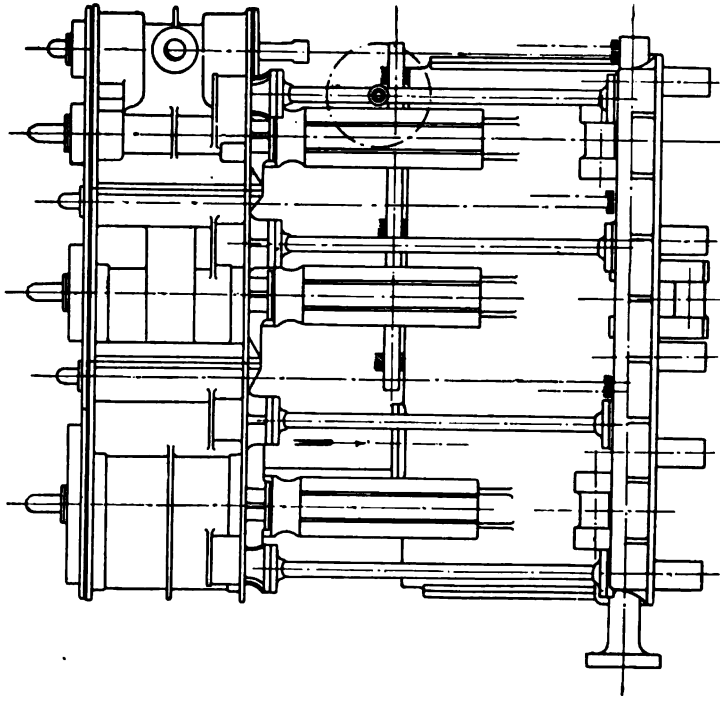

Fig. 82.

system of balancing is used, a good even turning moment is obtained;* and lastly, the strain on the crank shaft is not so great, because the pressure of the connecting rods is more evenly distributed over four, instead of over three crank pins. This arrangement is used in torpedo-boats and destroyers, warships of all sizes, mail and fast steamers.

Figs. 254, 255 (see page 252) show the engines of twin-screw destroyers built by Thornycroft & Co. Diameter of cylinders, H.P. 22 inches, M.P. 29 inches, L.P. 2 × 30 inches. Stroke, 18 inches. Boiler pressure above atmosphere, 225 lb. per square inch. Number of revolutions, 390. I.H.P. of each engine = 3,000. The engines are mounted on light

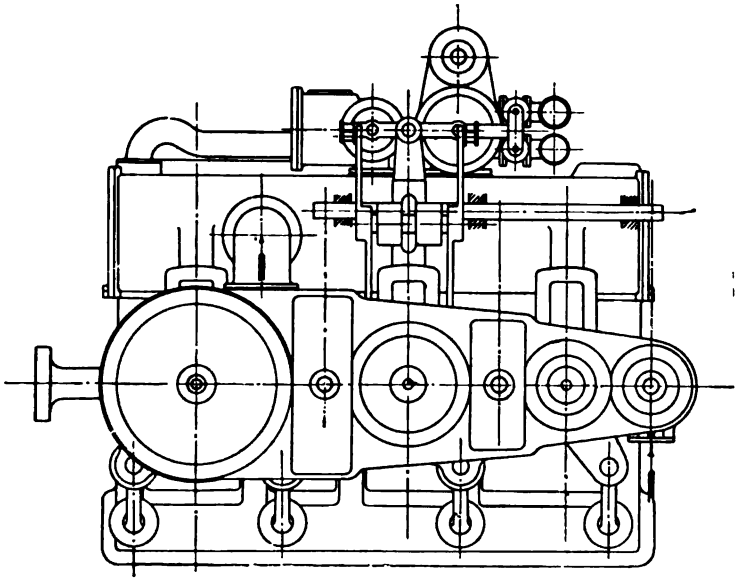

Fig. 84.

columns, the bed-plate is also made as light as possible, so that practically the engine foundations are formed by the engine seatings and framing of the ship.

Fig. 89, page 116, shows the engine of a destroyer with twin screws, built by Thornycroft for the Japanese Navy. Each engine is of 3,600 I.H.P., and runs at 390 revolutions per minute. Cylinder diameters, H.P. 22 inches, M.P. 30 inches, $L.P._1$ 31 inches, $L.P._2$ 31 inches. Stroke, 19 inches. Boiler pressure, 250 lb. per square inch. The forward pair of cranks is at 90° to the after pair, and the individual cranks of each pair are opposite each other. Compare Balance, page 90.

* See Lorenz, *Dynamik der Kurbelgetriebe*.

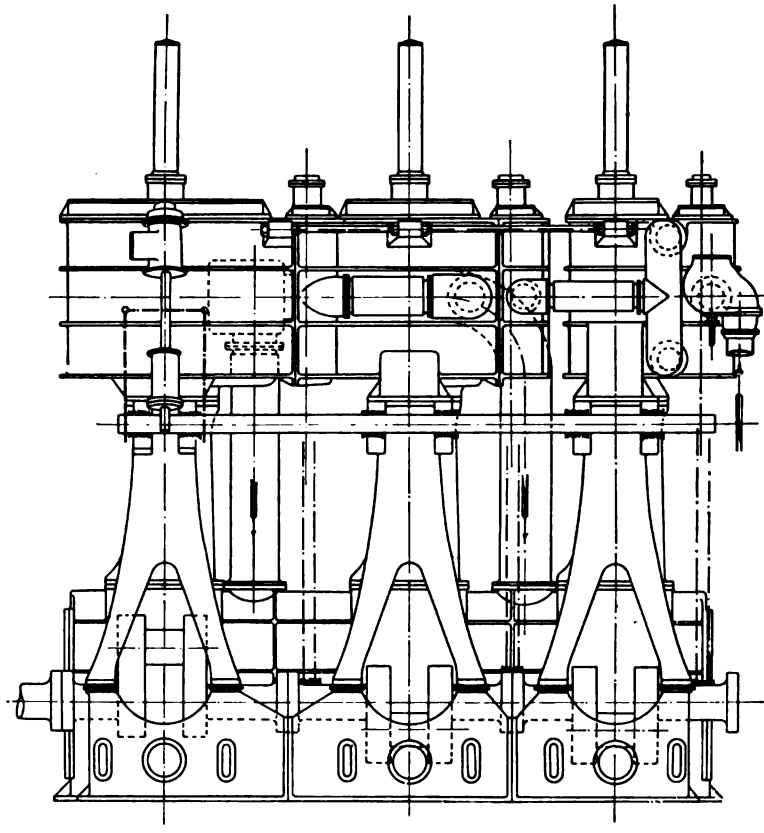

Fig. 85.

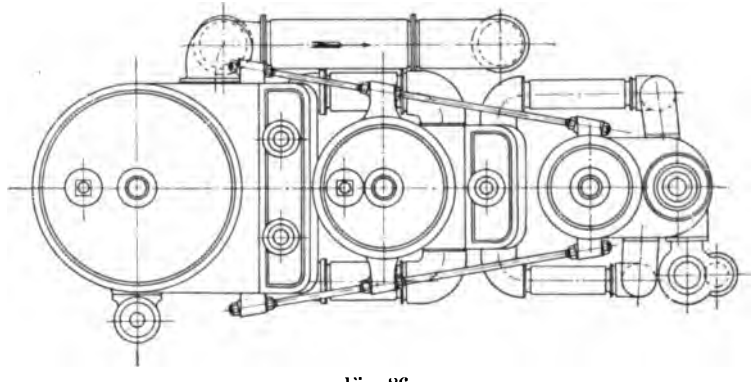

Fig. 86.

Fig. 113, page 126, belongs to the same type of engine. Fig. 90 (see Plate IV.) gives a similar engine for a small armoured cruiser with twin screws. I.H.P. $2 \times 3{,}750$, $n = 178$. Boiler pressure above atmosphere, 210 lb. per square inch. The two low-pressure cylinders have one valve chest between them. The cylinders are supported on steel columns in front, and on cast-steel framing at the back. Stephenson's link motion, separate condenser, and Schlick system of balancing.

Figs. 93, 94, 95, and the photographs Figs. 91 and 92, show one of the two engines of the Russian cruiser "Bogatyr," which was completed in the summer of 1902 at the Vulcan Works, Stettin.* The engines are built on the Schlick system, each of them developing 10,000 I.H.P. at a speed of 150 revolutions per minute. Dimensions :— Diameter of H.P. cylinder = 40 inches, M.P. cylinder = 60 inches, of both L.P. cylinders 70 inches. Stroke, 36 inches. Admission pressure in H.P. cylinder, 240 lb. per square inch. The columns and bed-plate are of cast steel. The L.P. cylinders have balanced slide valves. (Compare Fig. 153, page 162.) The reversing gear is an all-round gear with two cylinders $6\frac{1}{4}$ inches diameter × 5 inches stroke; the turning gear has a single cylinder of the same dimensions. Each engine has an independent air pump of the Weir type; each of these has two steam cylinders $12\frac{1}{2}$ inches diameter, two pump cylinders 37 inches diameter, and stroke 15 inches. Each engine has a circular condenser with a cooling surface of 10,760 square feet, and a circulating pump with two vane wheels of 47 inches external diameter.

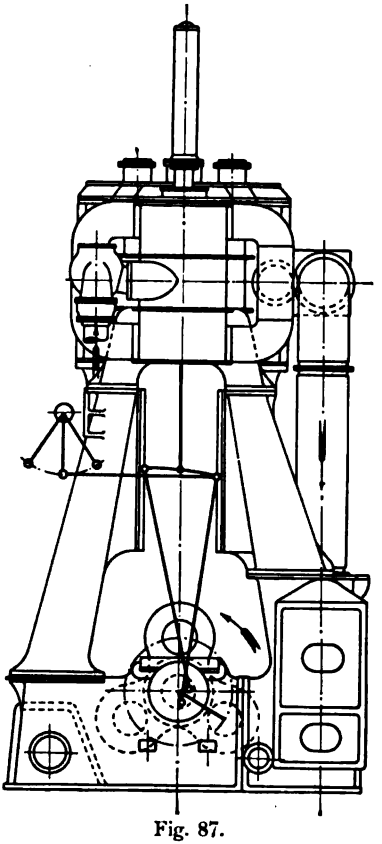

Fig. 87.

To a similar type of vessel belong the engines of the Japanese armoured cruiser "Yakumo," shown in Fig. 96, Plate V. At about 140

* Compare *Zeitschrift des Vereines Deutscher Ingénieure*, 1902, and *Engineering*.

revolutions per minute, each of the two engines indicates 8,000 H.P.; they are balanced on the Schlick system. The bed-plate is of cast steel, and each cylinder rests on four cast-steel ribbed columns. This arrangement gives increased stiffness, but at the sacrifice of accessibility and facility of inspection, as compared with the type described above.

Figs. 97 and 98, Plate VI., and Fig. 99, show the engines of the twin-screw fast steamer "Kaiser Wilhelm der Grosse," built at the Vulcan

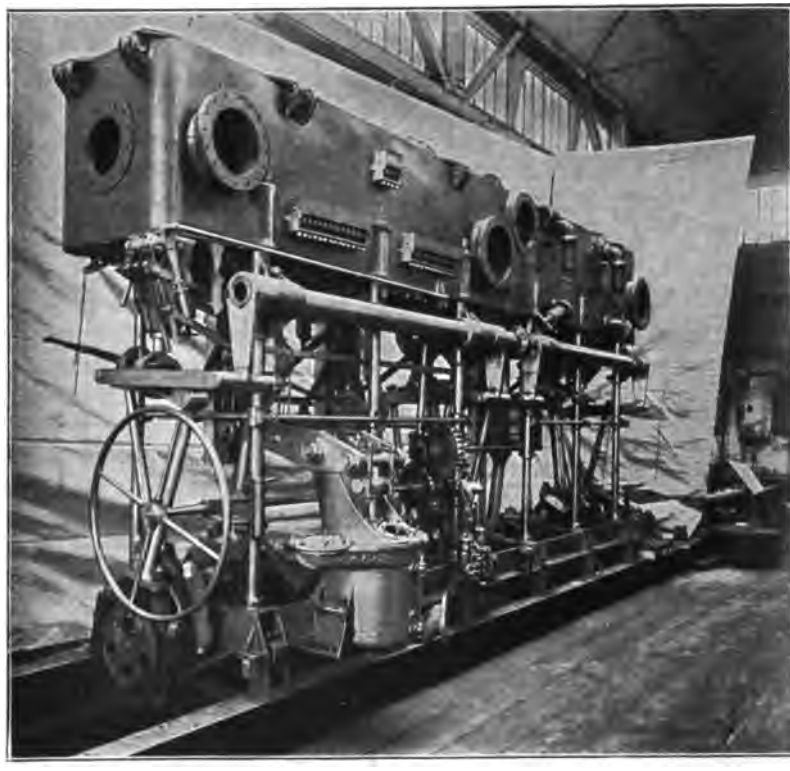

Fig. 89.

Works, Stettin, 1896. I.H.P. 2 × 14,000, $n = 78$. Diameter of cylinders, 52 inches, 90 inches, and two of 96 inches. Stroke, 69 inches. Boiler pressure above atmosphere, 210 lb. per square inch. The main engines are balanced on the Schlick system. Each cylinder is supported on four cast-steel ribbed columns; the bed-plate is of cast iron. The reversing gear is Stephenson's link motion. The H.P. cylinder has one piston valve, the M.P. cylinder two; the L.P. cylinders have flat slide valves. Each of the engines has a condenser of sheet copper with a cooling sur-

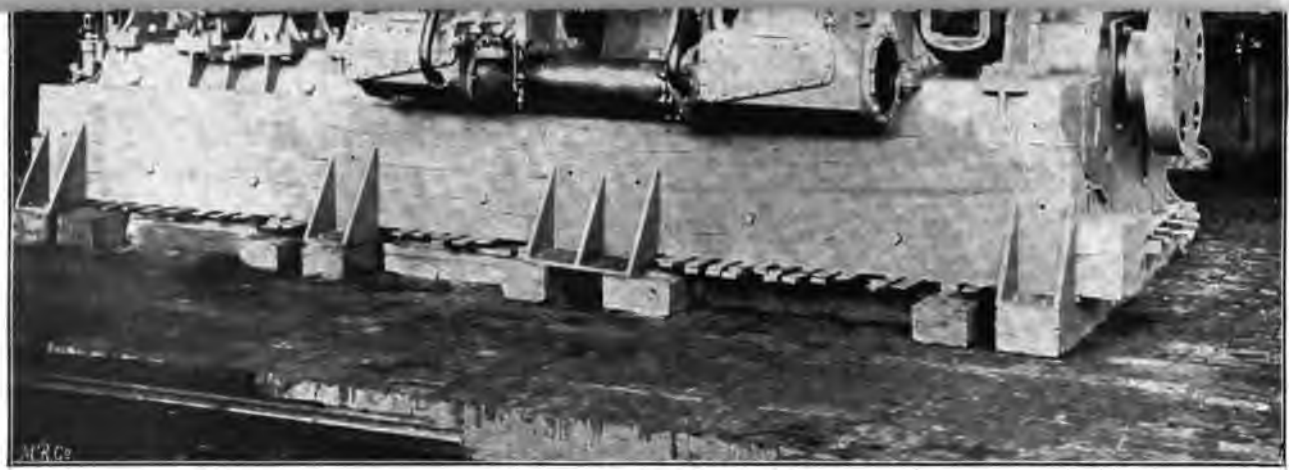

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Fig. 88.

Triple-expansion Engine of Imperial Yacht "Hohenzollern."

PLATE IV.

[*To face page* 116 (*after Plate III.*)

face of 17,750 square feet, and 5,530 tubes. Each of the two Blake air pumps has two steam cylinders 18 inches diameter, and two pump cylinders 42 inches diameter and 24 inches stroke. The circulating pump for each condenser is driven by a compound engine with H.P. cylinder 11 inches diameter, L.P. cylinder 20 inches diameter, and 12

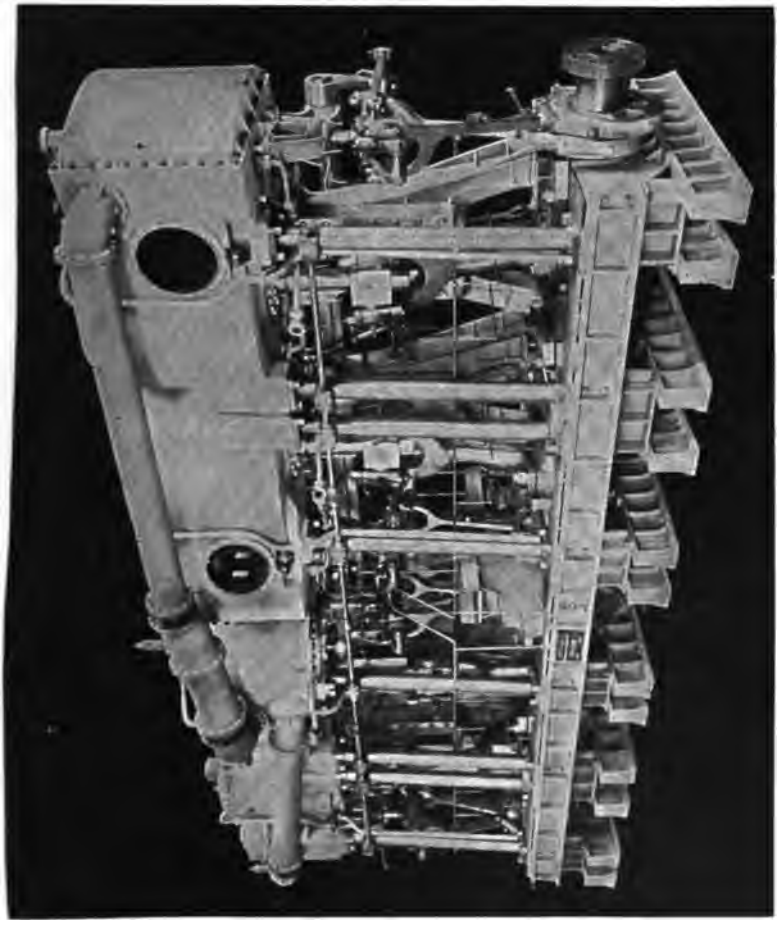


Fig. 91.

inches stroke. To either end of the shaft of the circulating engine is connected the vane wheel of a centrifugal pump, 48 inches external diameter and 20 inches diameter of suction, so that for the two main condensers there are in all four centrifugal pumps. Each main engine is fitted with a Brown's steam reversing gear, the steam cylinder of which is 26 inches diameter, the hydraulic cylinder 16 inches diameter, and

stroke 20 inches; also a two-cylinder turning engine, each cylinder being 8 inches diameter and 8 inches stroke.

Particulars may also be given of the engines of the twin-screw fast

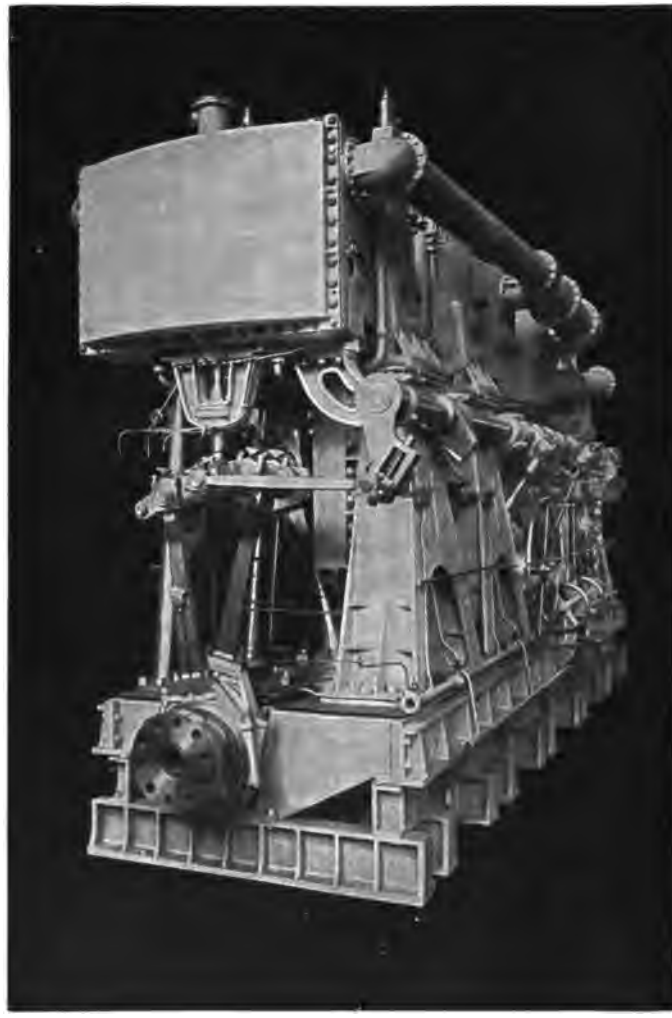


Fig. 92.

mail steamer, "Kaiserin Maria Theresa," built at the Vulcan Works, Stettin, in 1898. Each engine develops about 8,000 I.H.P. at 92 revolutions per minute. The cylindrical cast bronze condenser is separate

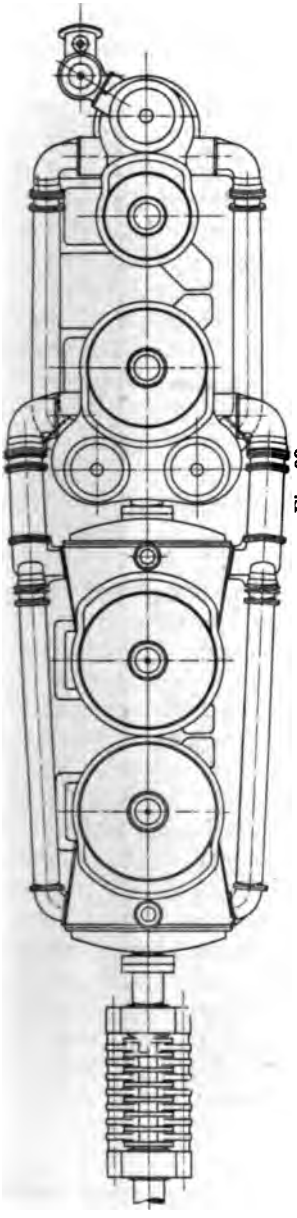


Fig. 93.

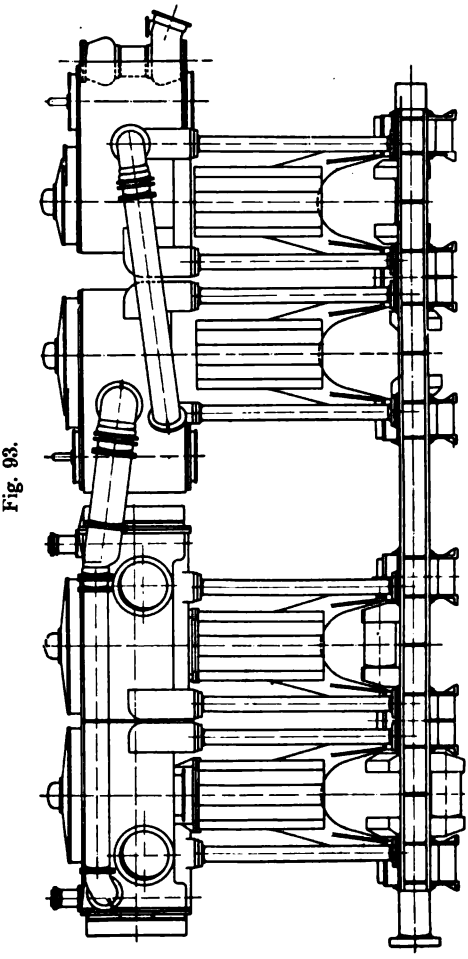


Fig. 95.

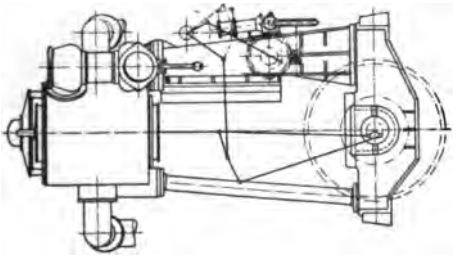


Fig. 94.

from the main engine. The latter has a bed-plate of cast iron, and is mounted on eight cast-iron columns of equal length. (See Fig. 100, Plate VII.) The engines are balanced on the Schlick system, and although they run at a high number of revolutions and high piston speed, there is not the slightest vibration in the hull of the vessel.

(*d.*) *Triple-expansion engines with two cranks and three cylinders* are now only used where local conditions do not allow of three cranks, as for instance when compound engines have to be tripled. The H.P. cylinder is then generally placed above the M.P. cylinder. The turning moment of such engines is always more irregular than that of three-crank engines.

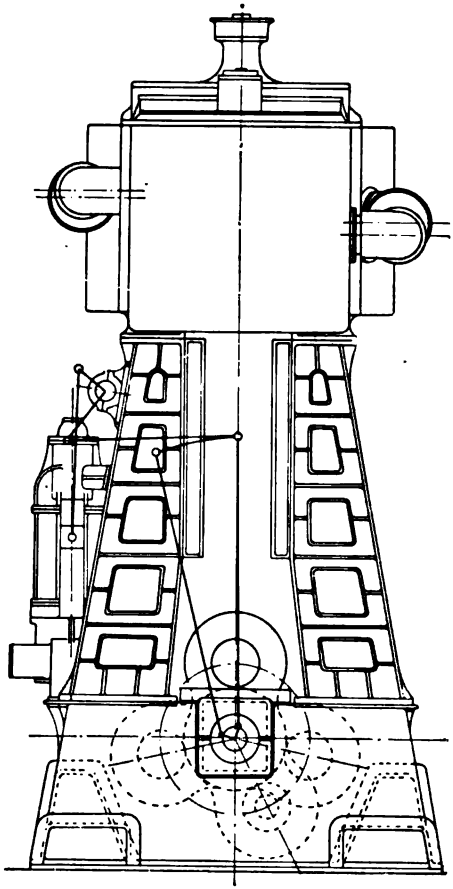


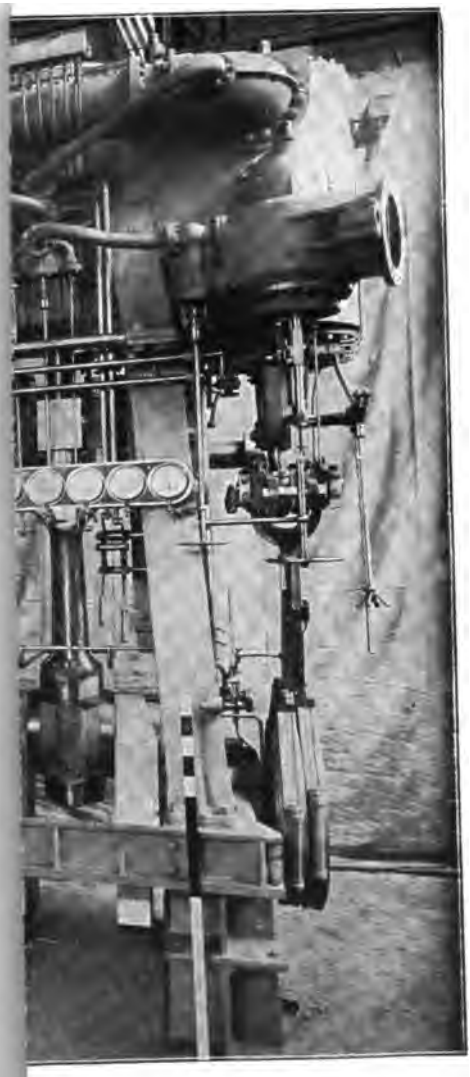
Fig. 99.

4. *Quadruple-expansion Engines.* — These have very varied arrangements of cylinders.

(*a.*) The most usual method is to place, the one H.P., the first M.P., the second M.P., and the one L.P. cylinder side by side, with the four cranks at an angle of 90°, corresponding to the Schlick system of balancing. The larger cylinders are generally in the middle. A similar engine is shown in Fig. 101, Plate VIII. It is that of a twin-screw Imperial Mail Steamer, each engine developing 4,500 I.H.P. Columns and bed-plate of cast iron; Stephenson's link motion. Balanced on the Schlick system. The second M.P. and the L.P. cylinders are in the middle, the H.P. cylinder forward, and the first M.P. aft.

(*b.*) With very large engines the L.P. cylinder must for reasons given above be divided. The engine is then built as a three-crank engine with five cylinders, or as a four-crank with six cylinders. An example of the latter arrangement is shown in the engines of the "Deutsch-

PLATE V.



Yakumo."

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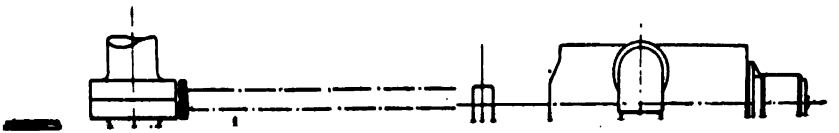


PLATE VII.

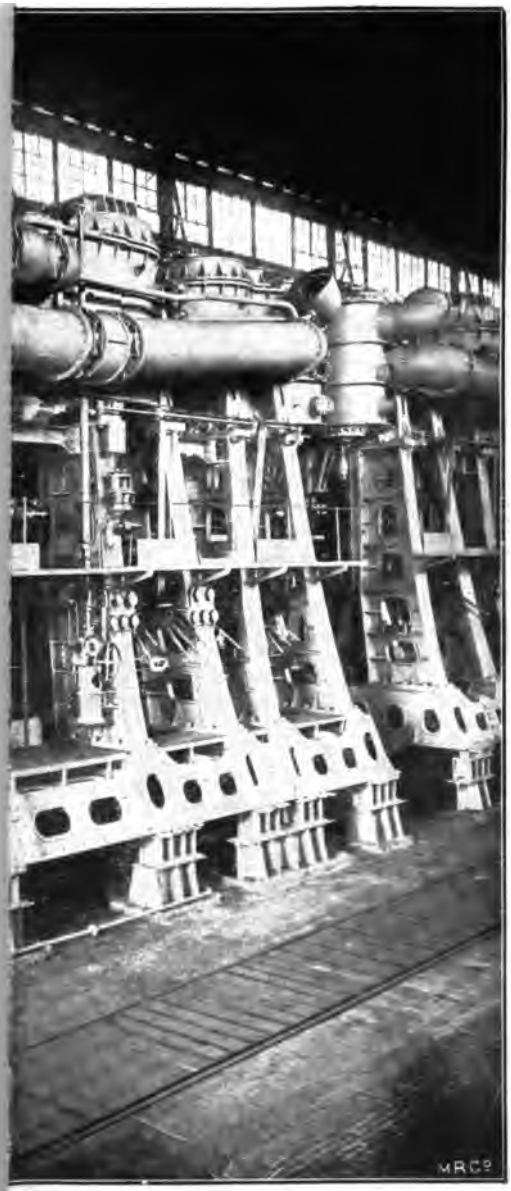
Maria Theresa."

[*To face page* 120 (*after Plate VI.*)

PLATE VIII.

[*To face page* 120 (*after Plate VII.*)

PLATE IX.



" Deutschland."

[*To face page* 120 (*after Plate VIII.*)

land" (Fig. 102, Plate IX.). I.H.P. = 2 × 17,500; revs. = 78. Diameter of cylinders, 2 × 930 mm. (two of 3 feet 6 inches), one 1,870 mm. (6 feet 1½ inches), one 2,640 mm. (8 feet 11 inches), and 2 × 2,700 mm. (two of 8 feet 10 inches); stroke, 1,850 mm. (6 feet 1 inch); boiler pressure, 214 lb. per square inch. Reversing gear, Stephenson's link motion.

Of the four cranks which are placed opposite each other, on the Schlick system, the foremost is driven by No. 1 M.P. cylinder. Then come the two L.P. cylinders, and No. 2 M.P. cylinder works the aftermost crank. An H.P. cylinder is mounted on the cover of each of the L.P. cylinders.* Each cylinder is supported on four cast-steel columns resting on a steel bed-plate. The two condensers are placed in the wings of the ship, and each has 5,320 condenser tubes, with 21,500 square feet of cooling surface. The air pumps, of which there is one to each condenser (and hence one to each engine), are Blake air pumps. Each has two steam cylinders 18 inches diameter, two pump cylinders 44 inches diameter × 24 inches stroke. Each condenser has also two circulating pumps, the vane wheels having an over-all diameter of 47 inches, and 30 inches diameter of suction on either side of the pump. The pumps are coupled direct to a compound engine. Diameter of H.P. cylinder of the latter 11 inches, of L.P. cylinder 20 inches × 12 inches stroke. The engines driving the turning and reversing gear are of the same dimensions as those of the "Kaiser Wilhelm der Grosse." The engines of the fast steamer "Kronprinz Wilhelm" are constructed on the same system as those of the "Deutschland."

(*c*.) The engines of the "Kaiser Wilhelm II.," also built at the Vulcan Works, are of a special type; the arrangements are shown in Fig. 105, and Figs. 106, 107, Plate X. Each of the two crank shafts has six cranks in all. The arrangement of the cylinders by which these cranks are driven is as follows, beginning from the after end:—First an L.P. cylinder, then No. 2 M.P. cylinder, next No. 1 M.P. cylinder with an H.P. cylinder on top, then No. 1 M.P. cylinder with an H.P. cylinder on top, next No. 2 M.P. cylinder, and lastly an L.P. cylinder. Thus the three after and the three forward cranks are each, as it were, driven from a quadruple-expansion engine. A watertight bulkhead divides the after from the forward three-crank engine. The sequence of the cranks is shown in Figs. 103, 104. The port and starboard engines together develop about 40,000 I.H.P. at 80 revolutions per minute, with admission pressure of 245 lb. per square inch above atmosphere. The dimensions are as follows:—

* In the photograph, Fig. 102, the two H.P. cylinders which are mounted on the covers of the L.P. cylinders, are not in place. For their arrangement see "Steam Cylinder" Section, and for complete description see *Zeitschrift des Vereines Deutscher Ingénieure*, 1900.

Diameter of H.P. cylinder, 38 inches. No. 1 M.P. cylinder, 50 inches.
„ No. 2 M.P. cylinder, 75 inches. L.P. „ 112 „
Stroke, 70 inches.
Diameter of crank shaft of the three forward cranks, 20 inches.
„ „ „ „ after „ 25 „
„ crank pin of all six cranks 25 „
Total length of crank shaft from the forward end to the aftermost flange coupling } 72 feet.

The condensers, of which there is one for each aggregate of three

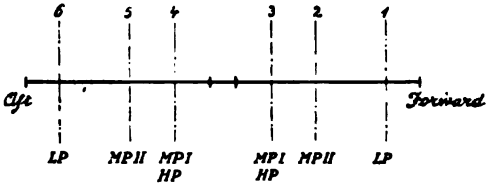


Fig. 103.

cranks, and therefore two for each of the port and starboard engines, have each a cooling surface of 11,700 square feet. Each condenser is fitted with an independent Duplex air pump, and a single-cylinder circulating pump. The port and starboard engines have each two Brown's reversing engines, the valves of which are connected by a rod, and worked with the greatest ease through a single hand lever. The crank shaft is of nickel steel. Each cylinder is supported on two hollow cast-steel columns with large openings in them. The bed plate consists of cast-steel cross girders, with flanges for bolting them together longitudinally.

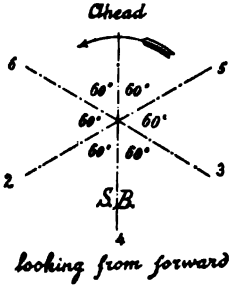


Fig. 104.

§ 49. **Longitudinal Bracing of the Cylinders.**—Most engineers endeavour so to connect the cylinders, that the whole engine forms one stiff structure. This is a necessary arrangement, to ensure the stability of each cylinder, and to prevent vibrations in the ship. It is obtained by the use of strong bed-plates and engine seatings, and also by bracing the cylinders or the upper ends of the frames together as firmly as possible. In connecting the cylinders of large engines, allowance must be made for expansion, *i.e.*, the tie-bars or connecting supports should be joined in such a way, that each cylinder can expand without throwing the one next to it out of line. (See Figs. 108 and 109.) The arrangement in

Fig. 108 is not so good as in Fig. 109, which allows for free expansion. Similarly the arrangement in Figs. 111 and 112 is preferable to that shown in Fig. 110.

In small engines the dimensions of the cylinders are less, and their expansion is not of so much importance, so that there is no disadvantage in having all the cylinders cast in one, or rigidly connected through their valve chests.

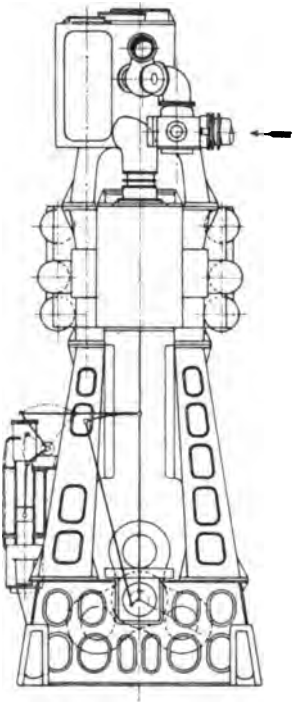


Fig. 105.

§ 50. **General Remarks on the Arrangement of the Main Engines.** —The arrangement of the cylinders depends in great measure on the choice of the valves. In triple-expansion engines piston valves are generally used for the H.P. and often for the M.P. cylinders, and flat slide valves for the L.P.; in quadruple-expansion engines the second M.P. cylinder is sometimes fitted with flat slide valves. For fear of excessive wear, large flat slide valves are generally avoided when the steam is at any considerable pressure, say 55 to 85 lb. per square inch. For large powers two piston valves or two flat slide valves are placed side by side in the same valve chest. If it is desired to make the engines very short, a valve gear is selected which will allow of the valves being placed at the side of the engine, such as the Heusinger or the Klug valve gear. (See pages 176, 181.) If the lengths of the crank shaft have to be the same, the distances between the cylinders are practically fixed, because only two long or two short sections, or one long and one short section can be coupled together. (See "Crank Shaft.") But this need not be considered in light engines for warships, the crank shafts of which are forged in one piece.

In balanced engines the valve chests are so placed that both the distances between the cylinders, and the position of the eccentrics, are favourable to balancing. (See § 35.) If the engines are not balanced the cylinders are sometimes crowded closely together, the object being to diminish the twisting of the engine in the longitudinal direction as much as possible.

To make the most of the available space, and especially to reduce

the height of the machinery, Thornycroft has built engines for torpedo-boats and destroyers, with inclined columns, the piston rods of which therefore are at acute angles to each other. This class of engine has been fitted in H.M. destroyers "Ardent," "Boxer," and "Bruiser."

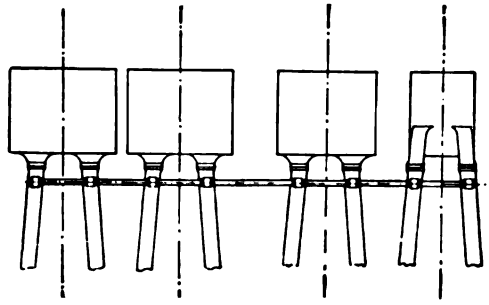


Fig. 108.

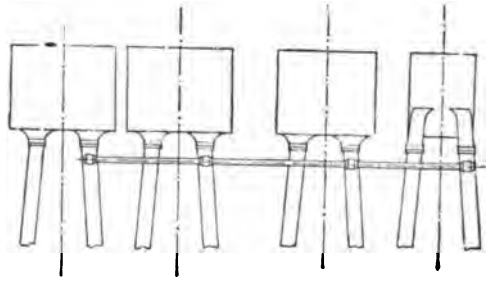


Fig. 109.

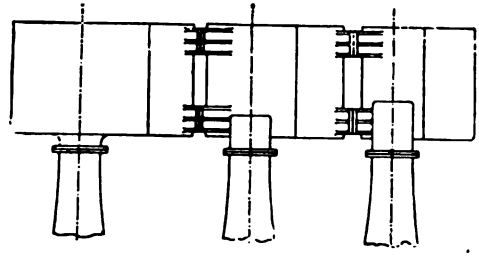
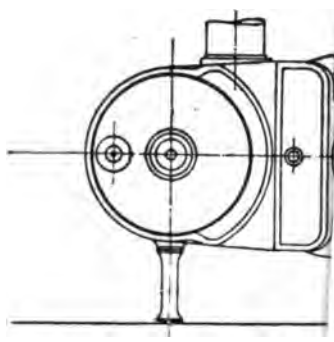
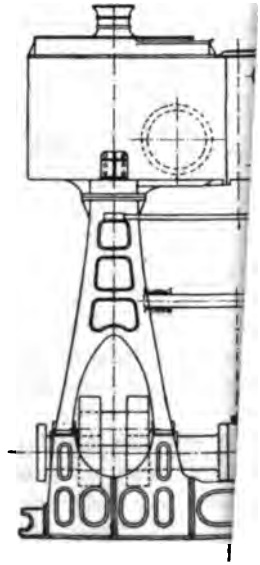


Fig. 110.

I.H.P. per engine 2,200, at 390 revolutions per minute. Diameter of cylinders, H.P. 19 inches, M.P. 27 inches, two L.P. 27 inches. Stroke 16 inches. Boiler pressure 200 lb. per square inch. The two forward cranks are at right angles to the two after cranks, and the individual cranks of



aft



each pair are at such an angle that if the cylinder axes were in the same plane, they would be at 180°. (Thus one piston of each set is always at the upper dead point when its fellow piston is at the lower dead point, and *vice versâ*.) Under certain conditions this type of engine is well balanced. (Compare page 90.) Although it is desirable, as far as balancing the engines is concerned, to place the large cylinders in the centre,* yet to facilitate the working and overhauling of the engines, it is better to place the casing containing the large flat slide valves at either end of the engine, so that the covers of the slide-valve chests can be easily taken off, and

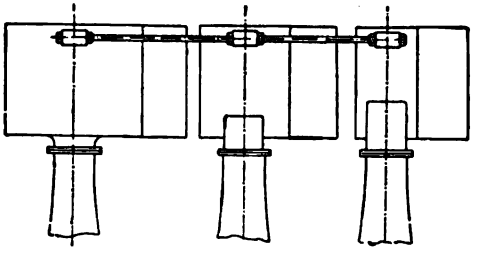


Fig. 111.

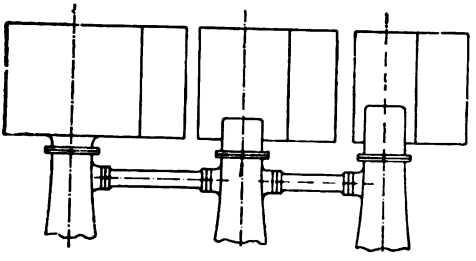


Fig. 112.

the valves removed for inspection. If a flat slide valve is between two cylinders, care must be taken to leave sufficient space between its cover and the adjacent cylinder so as to be able easily to remove the valve.

§ 51. **Starting the Engine.**—As all marine engines, and especially those for warships, must be rapidly manœuvred, it is necessary that they should be easily and quickly started. This must be taken into account when determining the position of the cranks. Nearly all marine engines have *starting* or *bye-pass valves* (see page 143) to

* In some engines with Schlick balancing the slide valves of the L.P. cylinder are quite outside, and the connecting rods of the outer cranks are consequently made as light as possible (see Fig. 114).

admit live steam into either the M.P. or the L.P. receiver, or to the top or bottom of the M.P. or L.P. cylinder. The diagram, Fig. 115, for a three-crank engine, shows where the auxiliary steam should be introduced. The cut-offs are marked in the concentric circles, of which the outermost represents the L.P., the middle the M.P., and the inner the

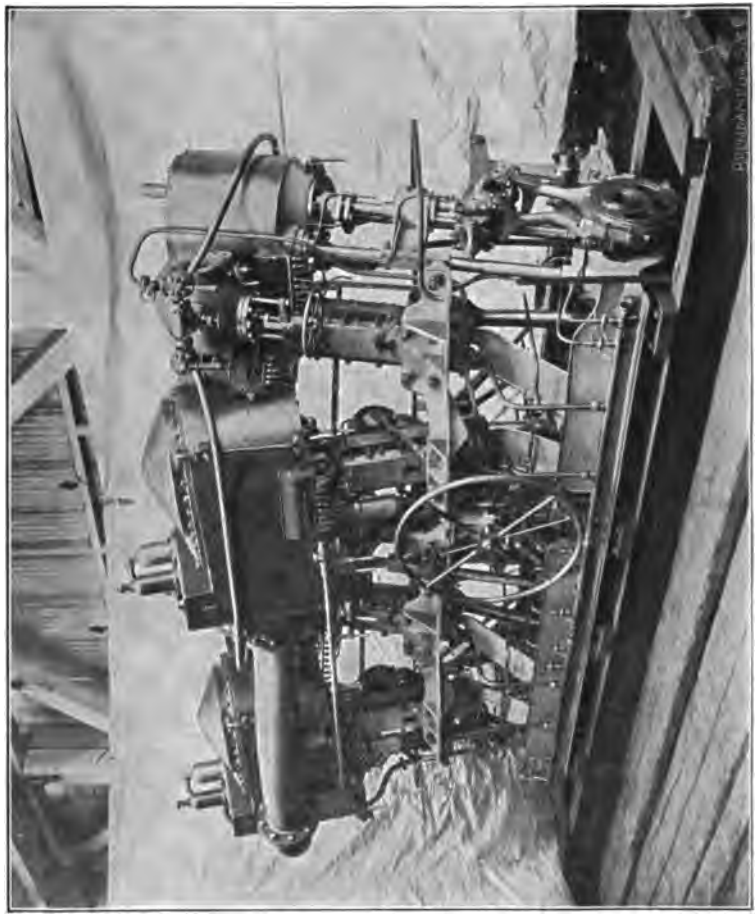


Fig. 113.

H.P. cylinder. The H.P., M.P., L.P. crank shaft is made to rotate on its axis through any position. If one of the cranks falls within the arc or section representing the cut-off of its corresponding cylinder, the engine can be started if live steam is admitted to that cylinder on the proper side. It must not be forgotten, however, that, up to about 20° beyond the dead point, the crank cannot exert sufficient turning moment to set

the engine in motion. The diagram shows at what positions of the crank the engine will not start, and to top or bottom of which cylinder, or to which receiver, the live steam should be supplied. If for any given position of the crank shaft, none of the cranks are in their

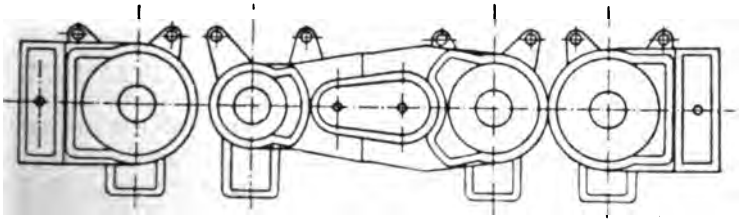


Fig. 114.

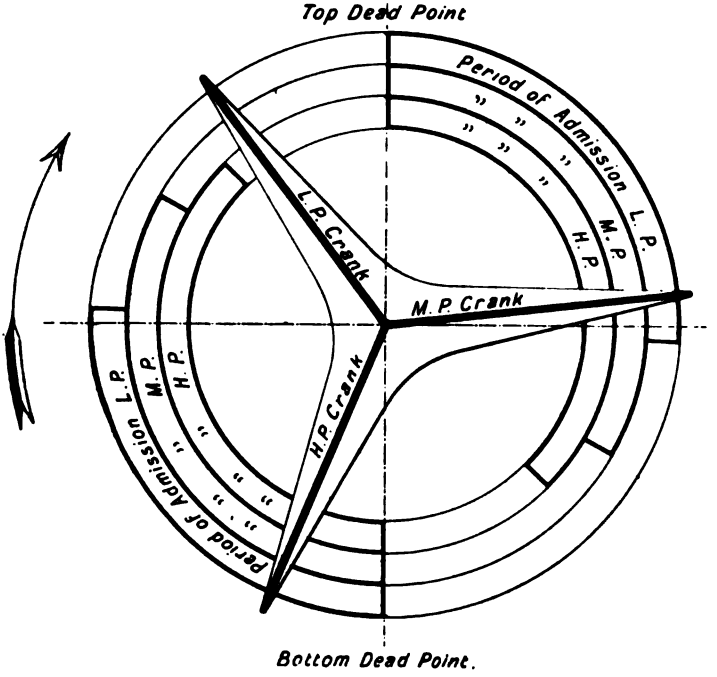


Fig. 115.

corresponding arcs representing the cut-offs, the engine will not start merely by supplying steam to the receiver. If live steam cannot be supplied to the top or bottom of a cylinder, the valve gear must be reversed, the engine started backwards, a better position of the crank obtained, and the engine re-started in the right direction.

SECTION V.

DETAILS OF MAIN ENGINES—THE CYLINDER.

§ 52. **General Remarks.**—Steam cylinders are almost universally made of the best fine-grained cast iron; gunmetal or bronze being only used in exceptional cases, and then only for small and specially light engines. If this latter metal be employed, it is usual to make the piston and piston rings of steel or cast iron, because bronze working upon bronze does not wear well.

The cylinders are made either with single or double walls. *Single or unjacketed walls* are used in the main engines of small warships, where it is necessary to have them extremely light, or in the engines of ordinary small or medium-sized cargo or passenger steamers, and in all auxiliary engines—in other words, wherever economy of weight, cheapness, and simplicity of construction have to be considered. *Double walls* are used either when steam jackets are used, or when the liners may need renewing.

§ 53. **Velocity of Steam.**—The mean velocity of the piston, corresponding to the normal speed of the engine, is always used as the basis on which to calculate the different cross-sectional areas of the passages, receiver pipes, &c., and is—

$$c = \frac{s \cdot n}{30} = \text{feet per second.}$$

Here c = piston speed in feet per second; s = stroke in feet; n = revolutions per minute.

The mean velocity of the steam (v) in ordinary engines is as follows:—

1. Main steam pipe. $v = 100$ to 130 feet per second. If the steam pipe is very long the speed should be a little less.

2. In steam passages of the H.P. cylinder, $v = 80$ to 100 feet per second; M.P. cylinder, $v = 100$ to 120 feet per second; L.P. cylinder, $v = 120$ to 140 feet per second.

3. In exhaust passages and receiver pipes of the H.P. cylinder, $v = 65$ to 80 feet per second; M.P. cylinder, $v = 80$ to 95 feet per second; L.P. cylinder, $v = 95$ to 110 feet per second.

In very light quick-running engines, where a saving of weight and space is of more importance than economy, the steam velocities given above may be increased by 10 to 20 %.

Table No. 16.

Cross Sectional Areas of Steam Passages.

Piston Speed.	Main Steam Pipe.	Steam Passages.			Exhaust Passages.		
		H.P. Cylinder.	M.P. Cylinder.	L.P. Cylinder.	H.P. Cylinder.	M.P. Cylinder.	L.P. Cylinder.
ft. per second.							
6·56	0·258 to 0·224	0·08 to 0·067	0·067 to 0·056	0·056 to 0·049	0·100 to 0·083	0·083 to 0·072	0·069 to 0·059
7·38	0·274 ,, 0·237	0·09 ,, 0·075	0·075 ,, 0·063	0·063 ,, 0·056	0·112 ,, 0·094	0·094 ,, 0·080	0·078 ,, 0·066
8·20	0·289 ,, 0·250	0·10 ,, 0·083	0·083 ,, 0·069	0·069 ,, 0·061	0·125 ,, 0·104	0·104 ,, 0·089	0·086 ,, 0·073
9·02	0·303 ,, 0·262	0·11 ,, 0·092	0·092 ,, 0·076	0·076 ,, 0·066	0·138 ,, 0·115	0·115 ,, 0·098	0·095 ,, 0·081
9·84	0·316 ,, 0·274	0·12 ,, 0·100	0·100 ,, 0·083	0·083 ,, 0·071	0·150 ,, 0·125	0·125 ,, 0·107	0·103 ,, 0·088
10·66	0·329 ,, 0·285	0·13 ,, 0·108	0·108 ,, 0·090	0·090 ,, 0·077	0·163 ,, 0·135	0·135 ,, 0·116	0·112 ,, 0·096
11·48	0·342 ,, 0·296	0·14 ,, 0·117	0·117 ,, 0·097	0·097 ,, 0·083	0·175 ,, 0·146	0·146 ,, 0·125	0·121 ,, 0·103
12·30	0·354 ,, 0·306	0·15 ,, 0·125	0·125 ,, 0·104	0·104 ,, 0·089	0·188 ,, 0·156	0·156 ,, 0·134	0·129 ,, 0·110
13·12	0·365 ,, 0·316	0·16 ,, 0·133	0·133 ,, 0·111	0·111 ,, 0·095	0·200 ,, 0·167	0·167 ,, 0·143	0·138 ,, 0·118
13·94	0·377 ,, 0·326	0·17 ,, 0·141	0·141 ,, 0·118	0·118 ,, 0·101	0·212 ,, 0·177	0·177 ,, 0·152	0·146 ,, 0·125
14·76	0·387 ,, 0·334	0·18 ,, 0·150	0·150 ,, 0·125	0·125 ,, 0·107	0·225 ,, 0·188	0·188 ,, 0·160	0·155 ,, 0·132
15·58	0·397 ,, 0·345	0·19 ,, 0·158	0·158 ,, 0·132	0·132 ,, 0·113	0·237 ,, 0·198	0·198 ,, 0·170	0·164 ,, 0·140
16·40	0·409 ,, 0·354	0·20 ,, 0·167	0·167 ,, 0·139	0·139 ,, 0·119	0·250 ,, 0·208	0·208 ,, 0·178	0·172 ,, 0·147
Speed of the steam—feet per second .	100 to 130	80 to 100	100 to 120	120 to 140	65 to 80	80 to 95	95 to 110

The cross-sectional area of the passages is found from the equation—

$$f = \text{area of the cylinder} \times \frac{\text{velocity of the piston}}{\text{velocity of the steam}}$$

In Table No. 16 the cross sectional areas of steam passages and the diameters of main steam pipe are given for varying piston speeds, assuming respectively the area of the cylinder and the diameter of the H.P. cylinder to be unity.

§ 54. **Thickness (δ) of the Cylinder Liner.**—The thickness of the cast-iron cylinder liner may be determined from the following empirical formula, which gives fairly correct results for vertical cylinders for steam pressures above 85 lb. per square inch—

$$\delta = \frac{d \cdot p}{5{,}125 + 10p} + \cdot 4 \text{ inch}$$

in which d = internal diameter of the cylinder in inches; p = pressure of steam in pounds (above atmosphere); δ = thickness of the cylinder liner in inches. Table No. 17 is calculated from this formula. The thicknesses of liner here given may be used for ordinary single-cylinder engines, and for the high-pressure cylinder of multiple-expansion engines for passenger and cargo steamers. The thickness of liners may be made the same for the M.P. and L.P. cylinders as for the H.P. cylinder, without taking their diameters or the steam pressures into account.

TABLE No. 17.

Thickness of the H.P. *Cylinder Liner in next larger* $\frac{1}{16}$ *inch.*

Diameter of the Cylinder.	For 150 lb. per square inch.	For 175 lb. per square inch.	For 200 lb. per square inch.
4 inches	$\frac{1}{2}$ inch	$\frac{1}{2}$ inch	$\frac{9}{16}$ inch
8 "	$\frac{9}{16}$ "	$\frac{5}{8}$ "	$\frac{11}{16}$ "
12 "	$\frac{5}{8}$ "	$\frac{3}{4}$ "	$\frac{3}{4}$ "
15 "	$\frac{3}{4}$ "	$\frac{13}{16}$ "	$\frac{7}{8}$ "
20 "	$\frac{7}{8}$ "	$\frac{15}{16}$ "	$\frac{15}{16}$ "
24 "	$\frac{15}{16}$ "	1 "	$1\frac{1}{8}$ "
27 "	1 "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "
32 "	$1\frac{1}{8}$ "	$1\frac{1}{4}$ "	$1\frac{5}{16}$ "
36 "	$1\frac{3}{16}$ "	$1\frac{5}{16}$ "	$1\frac{7}{16}$ "
39 "	$1\frac{1}{4}$ "	$1\frac{7}{16}$ "	$1\frac{9}{16}$ "
43 "	$1\frac{3}{8}$ "	$1\frac{9}{16}$ "	$1\frac{11}{16}$ "
48 "	$1\frac{7}{16}$ "	$1\frac{5}{8}$ "	$1\frac{13}{16}$ "
51 "	$1\frac{1}{2}$ "	$1\frac{3}{4}$ "	$1\frac{15}{16}$ "
55 "	$1\frac{5}{8}$ "	$1\frac{13}{16}$ "	2 "
59 "	$1\frac{11}{16}$ "	$1\frac{15}{16}$ "	$2\frac{1}{8}$ "

In light engines the thicknesses of the liners may be slightly less than those given above, and the same applies to their M.P. and L.P. cylinders, provided they are first tested under water pressure. If a margin has to be left for reboring, the thickness must be increased by about $\frac{1}{8}$ to $\frac{3}{16}$ inch. For liners of wrought steel the thickness in the table may be reduced 30 to 35 %.

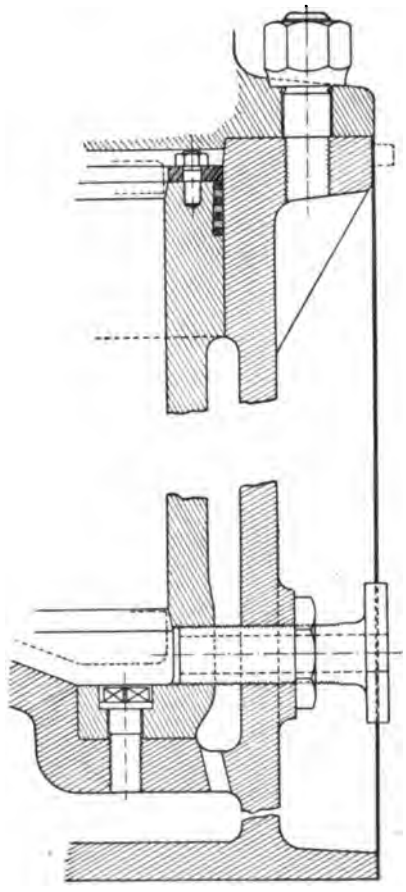


Fig. 116.

§ 55. **Thickness (δ_1) of the Cylinder Jacket or Outer Shell.**—The jacket is generally made of the same thickness as the corresponding liner; but the extra thickness of the liner, to allow for subsequent boring, need not be considered. The space between the jacket and the liner, if it is to be used as a steam jacket, must not be less than $\frac{3}{4}$ inch, and in large cylinders should be as much as $1\frac{3}{8}$ to $1\frac{1}{2}$ inch.

§ 56. **Thickness (δ_2) of the Walls of Cylinders without Liners.** — The walls of such cylinders are from 10 to 15 % thicker than those fitted with liners. To strengthen the cylinder, it is usually stiffened externally with circumferential rings or webs about $0{\cdot}8\delta_1$ in thickness, and rather less in depth than the cylinder cover flanges. These webs are usually spaced about ten to fifteen times the thickness of the cylinder walls apart.

§ 57. **Method of Fixing the Cylinder Liner.**—In small cylinders the liner is pressed into the cylinder casting and firmly secured at the top and bottom. The joint is made with asbestos cord at the bottom, and by a turned copper ring at the top, or by a kind of stuffing-box with asbestos and iron wire packing. Longitudinal movement is prevented by

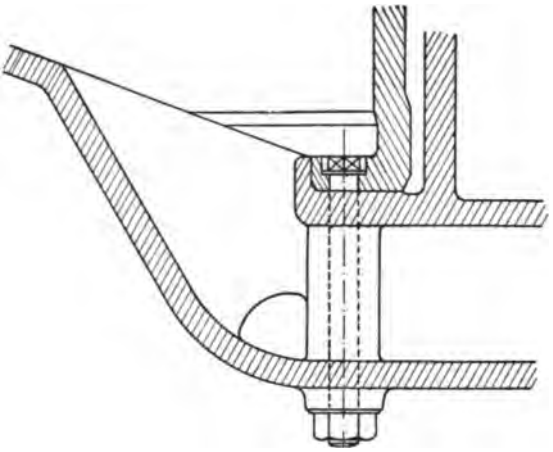


Fig. 117.

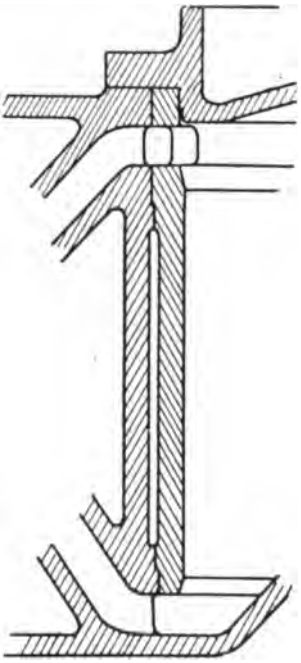


Fig. 118.

a few tap-bolts at the side, or by lengthening the liner till it touches the flange of the cylinder cover. (See Fig. 118.)

In large cylinders where more space is available, the cylinder liners are held to the bottom by a steam-tight flange and tap-bolts or studs. (See Figs. 116, 117, 119.) In engines with a longer stroke than about 3 feet, the liner has often a fitting ring in the middle, in which, if the jacket is heated, a few cross

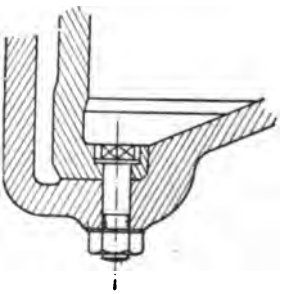


Fig. 119.

grooves are provided. The diameter d_1 of the tap-bolts or studs is about $= \delta$. The bolts, which are made with collars and squared heads,

are of the best wrought iron, or Siemens-Martin steel. Bronze screws are not to be recommended. (See Part VII.)

The *width of the holding-down flanges* should be as small as possible, and their thickness a about $= 1{\cdot}3\delta$, and the *spacing t of the bolts* is about—

$t = 4d_1$ for the H.P. cylinder
$t = 5{\cdot}5d_1$ for the M.P. „
$t = 7d_1$ for the L.P. „

and where possible they are made of the same size for all the cylinders.

The joint under the liner flange is best made with red lead and a thin copper wire, while the upper end is made steam tight with a copper ring well caulked into a slightly dovetailed groove. Asbestos and iron wire may also be used for this purpose instead of a copper ring. The joint is generally still further secured by means of a wrought-iron ring, held in place by $\frac{1}{2}$ to $\frac{5}{8}$ inch set screws. (See Fig. 116.) To avoid all shoulders on the wearing surface, the length of the actual working surface of the cylinder or liner is such that the piston rings are either flush with it, or overrun it by about $\frac{1}{2}$ inch at the top and $\frac{1}{4}$ inch at the bottom, according to their width. At each end, beyond the working surface of the liner, the metal should be cut or bevelled away, its diameter being increased by $\frac{1}{4}$ to $\frac{1}{2}$ inch. (See Fig. 116.)

§ 58. **Cylinder Cover Studs.**—As a rule the studs for securing the cylinder cover are made of the best wrought iron or steel. Their overall diameter d_2 may be approximately taken as about $= \delta$, or up to $1{\cdot}25\delta$. The stress on the screws, produced by the maximum steam pressure on the cover of the H.P. cylinder, should never be more than from 4,700 to 6,500 lb. per square inch. The spacing t of the studs varies with the steam pressure, and should be as follows:—

For pressures up to 50 lb., t = from $5d_2$ to $6d_2$.
„ „ from 50 to 100 „ t = „ $4{\cdot}5d_2$ „ $5d_2$.
„ „ „ 100 „ 170 „ t = „ $3{\cdot}5d_2$ „ $4{\cdot}5d_2$.
„ „ „ 170 „ 200 „ t = „ $2{\cdot}7d_2$ „ $3{\cdot}5d_2$.

It is desirable to make the studs the same size for the covers of all the cylinders, and their nuts should be about $1{\cdot}3d_2$ to $1{\cdot}5d_2$ deep, so that the thread of the screw may not wear out unduly quickly with the constant screwing and unscrewing which is apt to take place when shorter nuts are used. In the best work the nuts are case-hardened.

§ 59. **Cylinder Flanges.**—The thickness of the flange should be about $1{\cdot}3\delta_1$ to $1{\cdot}4\delta_1$, and the width $2{\cdot}6d_2$ to $3{\cdot}3d_2$. To strengthen the

flange it is desirable, in large cylinders, to fit radial ribs between each pair of bolt holes.

§ 60. **Cylinder Bottom.**—If cast-iron pistons are used, the bottom of the cylinder is generally flat, and must therefore be well stiffened with ribs, which serve at the same time to brace the supporting brackets or feet to the bottom and wall of the cylinder. The thickness of the bottom should be about $= \delta$ or $= 1{\cdot}1\delta$, and that of the ribs $= \delta$; the depth of the ribs should be at least 5δ to 6δ. The ribs should be as many in number as are required to afford sufficient support to the flat surfaces between them. If the cylinder bottom is double, the thickness of each wall should be $0{\cdot}9\delta$, and the distance between them, measured inside, about 5δ to 6δ, while the radial ribs bracing top and bottom together should be $0{\cdot}8\delta$ to $0{\cdot}85\delta$ thick. If conical pistons are used, the thickness of the walls may be from 0·9 to 0·95 of that given above. It is of special importance to make the circular centre rib, which forms the opening for both the piston rod and the boring bar, very strong.

§ 61. **Cylinder Feet.**—To connect the cylinders with the **A** frames or columns, they are cast with feet or faced brackets. In the design of these feet care must be taken that the strains coming upon them are distributed over as large an area of the cylinder shell or bottom as possible. The thickness of metal of the feet, if they are cast hollow or ribbed, is generally about $0{\cdot}85\delta$, and that of the flanges $1{\cdot}5\delta$ to $1{\cdot}6\delta$. The bolts securing the feet to the columns are of such dimensions that, at the maximum pressure on the piston, the stress on them will not be more than from 3,000 to 6,000 lb. per square inch. Their diameter being taken at $1{\cdot}5\delta$ to $1{\cdot}8\delta$, the number required can be easily determined.

§ 62. **Cylinder Passages and Valve Casings.**—If piston valves are used, the valve casings are cast in one with the cylinder, and are, where possible, cylindrical in shape. In small cylinders the valve casings are cast in one piece with the cylinder, but with larger cylinders for merchant vessels they are often cast separately, and fitted with an independent cover, so that the valve may be easily removed. In warships, to economise weight, and because the valve cannot be taken out from above, on account of the armoured deck, the valve chest is frequently cast in one with the cylinder, and is provided with a large cover at the side.

Thickness of valve-chest walls, $0{\cdot}9$ to $0{\cdot}95\delta$.

§ 63. **Calculation of Flat Surfaces in Valve Chests, Passages, and Covers.**—The flat walls of the valve chest and cover

must be stiffened with ribs to afford sufficient strength to the surfaces between them.

1. *Empirical Determination of Thickness of Walls and Pitch of Ribs.*—A flat cast-iron surface is strong enough if—

$$\delta = \frac{b}{113{\cdot}42\sqrt{p}}$$

where δ = thickness of the surface in inches; b = smallest distance between two contiguous ribs in inches; p = steam pressure in pounds per square inch (above atmosphere) on the surface in question.

Generally speaking, however, the thickness of the walls of the surfaces, between the ribs, is made *equal to the thickness of the cylinder liner itself.* The stiffening ribs must be so calculated that they will safely bear the steam pressures upon the surfaces A × B (see Fig. 121). B is here the smaller side of the rectangular area; the bending stress may be taken at about 4,500 lb. per square inch. About 6δ should be allowed, on either side of the rib, for the width of the flange of the T-shaped cross section. The height of the ribs should be about 4 to 5δ; thickness the same as that of the flat wall; and the distance of the transverse ribs from each other is

= about	12 to 14δ	for pressures of	50 lb.
„	10 „ 12δ	„	50 to 100 „
„	8 „ 10δ	„	100 „ 160 „
„	7 „ 8δ	„	160 „ 200 „

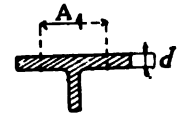


Fig. 120.

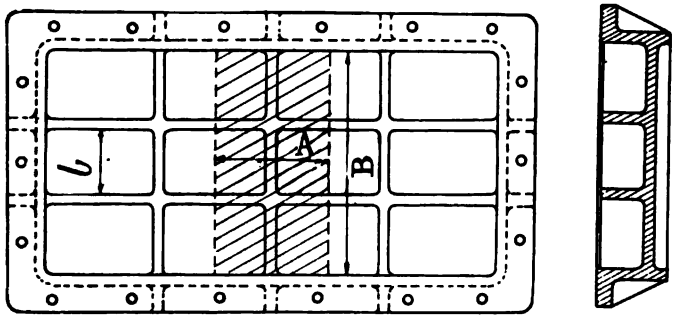


Fig. 121.

The ribs should, if possible, be placed on the side exposed to the pressure, as the metal is then better distributed to withstand the pressure than if they were on the other side, the larger area of the metal being thus in tension. Such ribs certainly offer considerable resistance to

the incoming steam, especially if they are transverse to its direction of entry, and they should therefore be allowed for, when determining the cross sectional area of the steam passages.

2. *Calculation of Flat Surfaces in Valve Chest, Passages, and Covers, on Bach's Method.*—From a large number of experiments it has been found that plates having corners, split across the diagonal line when subjected, under hydraulic pressure, to a uniform strain. Such covers should therefore be calculated for a bending stress along the diagonal. (See Fig. 122.)

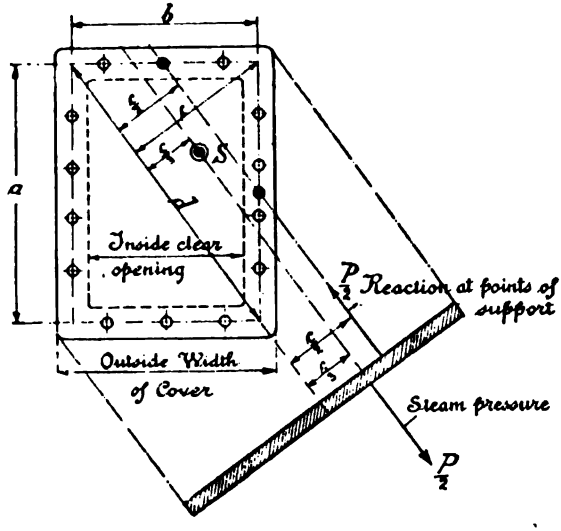


Fig. 122.

Load on the cover, $P = a \times b \times p$, p being the pressure per unit of surface. Half of the force P may be considered as acting at the centre of gravity s of half the cover; then the bending moment about the diagonal is $\frac{P}{2} \times \frac{c}{3}$. This force is counteracted by two forces which act along the sides a and b. The resultant moment of these forces about the diagonal is—

$$M_b = \frac{P}{2} \times \frac{c}{2} - \frac{P}{2} \times \frac{c}{3} = \frac{Pc}{12}$$

If c be expressed in terms of the sides a and b, and $P = a \times b \times p$, then we get—

$$M_b = \frac{P \cdot c}{12} = \phi \frac{p \cdot a^2 b^2}{12 \sqrt{a^2 + b^2}}$$

The coefficient ϕ is introduced to allow for the initial pressure; it varies from 1·125 to 0·75, but is generally taken as = 1.

The maximum stress in the cover is obtained from the equation—

$$\text{Bending stress} = \frac{\text{bending moment}}{\text{modulus of cross section}}$$

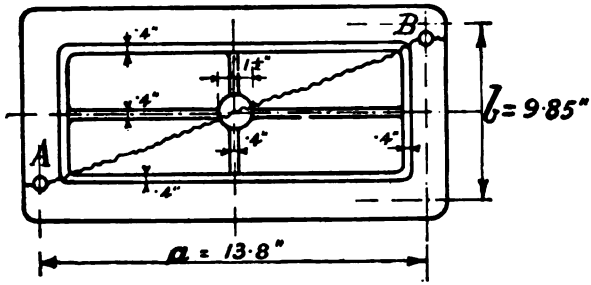


Fig. 123.

Example.—What is the stress in a cover, of the form shown in Figs. 123, 124? Working pressure, 185 lb. per square inch. Material, cast steel.

The moment of inertia of the section of this cover on the line AB is $I = 1{\cdot}478$ inches^4. Distance of the centre of gravity from the outside fibre is $e = 1{\cdot}09$ inch.

Therefore the modulus of cross section is $z = \frac{I}{e} = 1{\cdot}35$ inches^3.

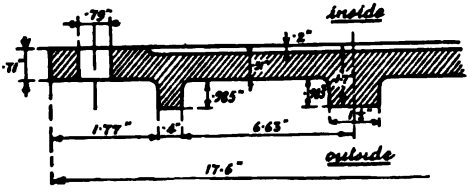


Fig. 124.

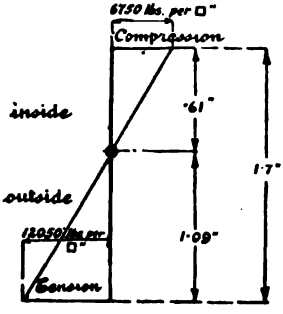


Fig. 125.

The bending moment is—

$$M_b = \frac{p}{12} \times \frac{a^2b^2}{\sqrt{a^2+b^2}} = \frac{185}{12} \times \frac{13{\cdot}8^2 \times 9{\cdot}8^2}{\sqrt{13{\cdot}8^2+9{\cdot}8^2}} = 16{,}850 \text{ in. lb.}$$

The stress in the cover is thus—

$$s = \frac{M_b}{w} = \frac{16{,}850}{1{\cdot}35} = 12{,}480 \text{ lb. per square inch.}$$

The diagram, Fig. 125, shows the distribution of the stresses.

§ 64. **Piston-valve Liner and Ports.**—In smaller engines a piston valve is often allowed to run direct in the bored casting of the valve chest, and a flat slide valve to work direct upon a valve face cast in one with the cylinder. In larger cylinders the piston valves work in specially hard cast-iron liners, fitted into the valve chests. (For dimensions see § 73.) These liners have to be accurately fitted and firmly bolted in position. (See Fig. 126.) In the same way for flat slide valves of larger cylinders separate valve faces of hard cast iron are screwed on to the cylinder faces so that they can be renewed when necessary. (See Fig. 127.)

The valve faces, when cast with the cylinder, are made about 0·9 to 1·1δ in thickness. The removable valve faces are made from δ to 1·1δ thick, and are fastened to the cylinder by wrought-iron or steel set screws, with round or squared countersunk heads, or, in small cylinders, with

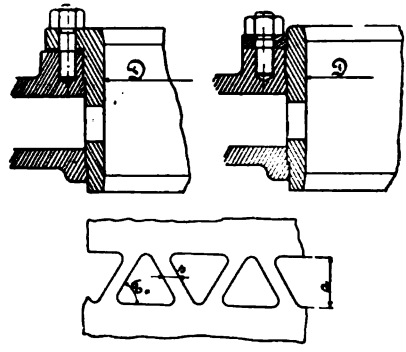


Fig. 126.

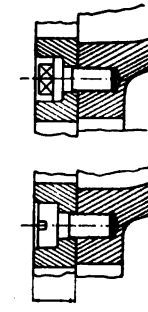


Fig. 127.

cheese heads. The pitch of these screws is usually about six to nine times their diameter. On account of the wear on the valve face, the heads of the screws should be sunk to such a depth that their upper edge is $\frac{1}{8}$ inch or so below the valve face. (See Fig. 127.) The liners and valve faces should be of the best hard close-grained cast iron.

§ 65. **Water Tests for Cylinders.**—Before the steam cylinders are covered with a non-conducting material, they should be tested by water pressure for strength and tightness. The pressures used in these tests, where p = boiler pressure in pounds per square inch, are—

1. For compound engines, H.P. cylinder about 1·3p.
   " " L.P. " 0·45p.
2. For triple-expansion engines, H.P. cylinder about 1·3 to 1·4p.
   " " " M.P. " 0·7 to 0·9p.
   " " " L.P. " 0·25 to 0·3p.

3. For quadruple-expansion engines, H.P. cylinder about 1·3 to 1·4p.

,,	,,	,,	1st M.P.	,,	0·75 to 0·9p.
,,	,,	,,	2nd M.P.	,,	0·4 to 0·5p.
,,	,,	,,	L.P.	,,	0·2p.

If the cylinder has a liner, the space between it and the shell must also be similarly tested by hydraulic pressure. The valve chest and cover are tested with the cylinder, but the exhaust passages are often submitted to a separate test.

§ 66. **Rules for Construction.**—Care should be taken that no undue local thickening occurs in the material of which the cylinder is made, as such places almost always become porous and leak. The ribs in the cylinder passages sometimes crack as the metal cools down after casting, and it is desirable to strengthen those which are exposed to strains by stay-bolts. Large cylinders are thus frequently strengthened by stay-bolts and ties at dangerous places. The opening in the cylinder bottom for the separate stuffing-box casting, or the hole for the piston rod, must be made of such a size that the boring bar can be passed through it. To join two cylinders by steam-tight flanges is only customary in small or medium-sized engines; with large cylinders it should be avoided. These should either be made quite independent of each other, or if bolted together, the connection should not be exposed to steam pressure. (See page 124.) The covers of the valve chests should be so arranged that the valves are easily accessible. In the same way care should be taken that the stuffing-boxes can be examined without difficulty while the engine is running. All flanges for connecting pipes and external fittings must be outside the *cylinder lagging*. The latter generally consists of fossil meal (a mixture of infusorial earth, asbestos fibre, and some binding or cementing material) for the H.P. cylinder, and felt with layers of asbestos, or felt only, for the L.P. cylinder. This is covered with sheet iron from about 20 to 14 S.W.G. in thickness, and should be as simple as possible, as elaborate sheet-metal lagging is expensive.

§ 67. **Cylinder Fittings.**—These are—

1. Regulating valve.
2. Throttle valve and governor.
3. Starting valve.
4. Cylinder and valve casing drain cocks.
5. Relief valves.
6. Indicator connections.
7. Connections and fittings for steam to jackets.

1. *Regulating Valve.*—This is fitted directly to the H.P. cylinder,

and is so arranged that the engineer can open and shut it conveniently and quickly from the platform by a hand lever or wheel.

Single-seated valves are as a rule only used in small engines, as when fully open they do not close quickly enough. To close large valves of this kind, when the steam enters below the seat, and to open them when the steam pressure is above the seat, requires too much power, and such valves are therefore only used when specially balanced. (See Fig. 128.)

In this type of valve the top of the valve is shaped like a piston and fits into a cylinder. There is a small valve and seating in the middle of the large valve which, when the valve is shut, is closed by the spindle. The steam first enters the outer part of the valve box, and as

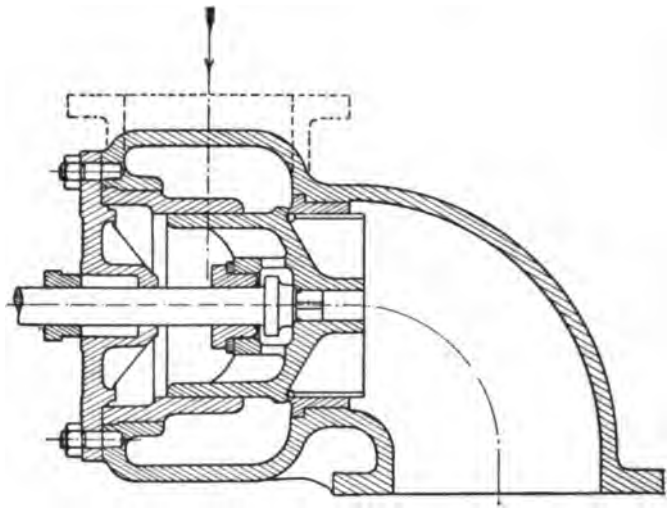


Fig. 128.

the piston does not fit tightly, it finds its way into the space above the valve and presses the latter firmly against its seat. As the spindle rises, it opens the smaller valve, which has a lift of about $\frac{1}{16}$ to $\frac{1}{8}$ inch, the steam above it escapes, and the large valve is then easily raised by the spindle. To avoid any chattering of the valve when it is open, the spindle is often fixed to it, in which case there is a small separate bye-pass valve on the valve cover, which can also be worked by the engineer from the platform. Through this bye-pass valve, the steam above the larger valve can pass into the H.P. cylinder and thus relieve the main valve. The spindle of the main valve is worked by the engineer from the starting platform either directly or by means of gearing, or sometimes by means of a special auxiliary engine.

Butterfly valves, instead of mushroom valves, are often used in large engines developing up to 2,000 H.P. These are also worked by hand levers from the engineer's platform. The angle of travel of the valve is about 90°.

In large engines *double-seated valves* are in favour, worked either direct or through an auxiliary cylinder, as shown in Figs. 129 to 131. The valve body, valve, separately fitted valve seat, and the spindle are of bronze. The upper valve is only from $\frac{1}{8}$ to $\frac{1}{4}$ inch larger in diameter than the lower one, so that both valves can be drawn out at the top, and at the same time the difference of pressure on the two valves is reduced to a minimum. The valve spindle is carried downwards through a stuffing-box, and has a rod for operating the valve attached to its lower end. When the engine is completely shut down the valve can be closed by means of a hand wheel and screw acting on the upper end of the spindle, which projects through the top cover of the valve casing. The main steam pipe is connected through an expansion joint forming part of the stop valve. (See "Main Steam Piping," page 423.) A throttle valve is frequently placed either outside this valve, or between it and the H.P. cylinder. This can be operated by the engineer either by hand or by a governor (of the Aspinall or some similar type), to prevent the racing of the engine in rough weather.

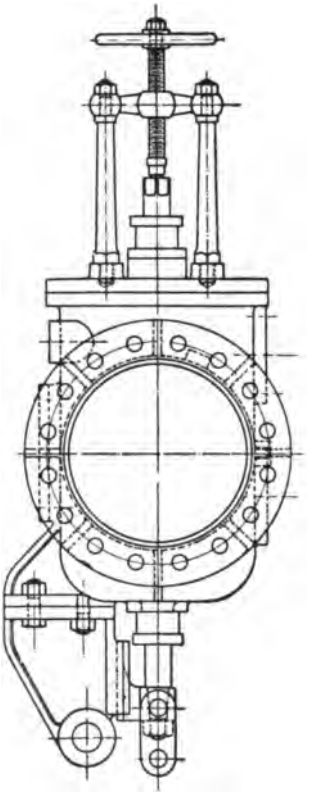


Fig. 129.

2. Mention may here be made of the *Aspinall Governor* (Figs. 132, 133, 134, Plate XI.). It consists of a weight W, hinged to a bracket, which acts upon two pawls P, P_1, and is carried on a frame, bolted to the air-pump lever or some other reciprocating part. If the normal speed of the engine is exceeded by about 5 %, the weight W, in conformity with the laws of inertia, lags behind on the downward stroke of the governor, and is held fast by a clutch, causing the lower pawl P to project, and the upper pawl to be brought back. When the governor reaches its lowest position, the lower pawl catches under lever H, and carries it with it up to the highest position, thus shutting off steam by closing the throttle valve. On the return stroke of the governor, the

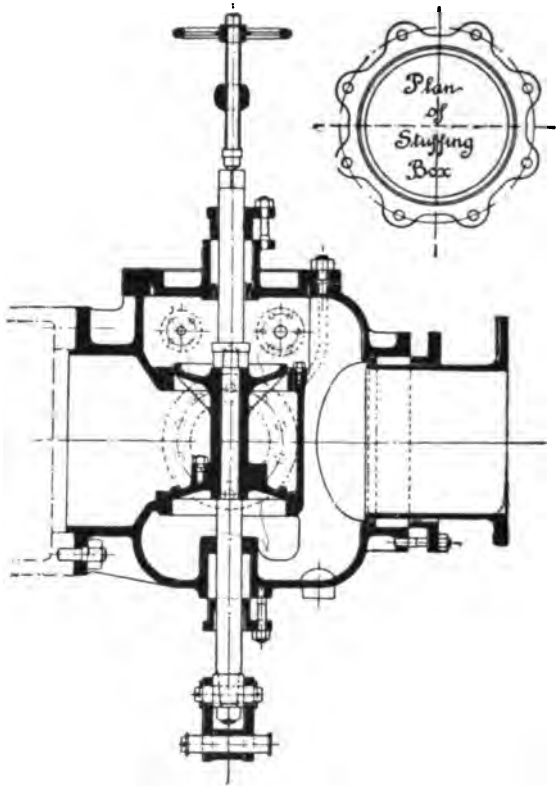


Fig. 130.

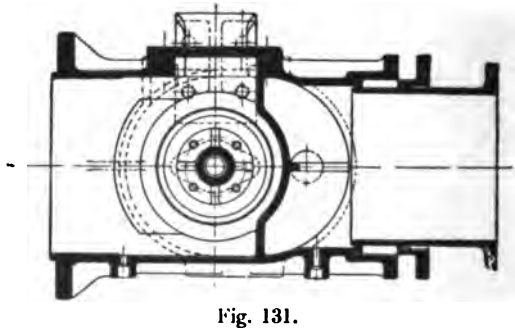


Fig. 131.

latch D strikes against lever H, after the upper of the two pawls has already slipped past it; W is released, and the two pawls again take up their normal position, the upper one in front, the lower behind. So

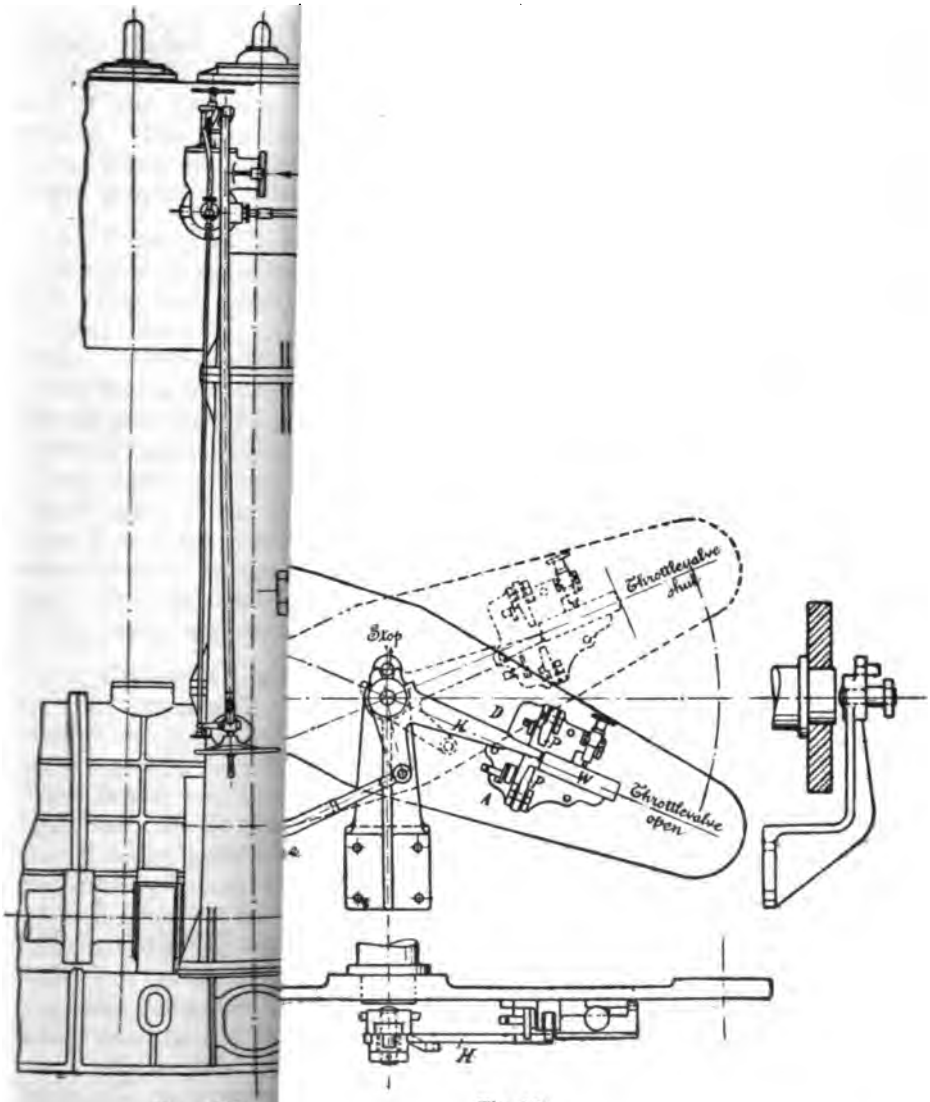


Fig. 132.

Fig. 134.

[*To face page* 142.

long as the speed of the engine exceeds the normal, this action is repeated when the governor is in its highest position, and the throttle valve is held closed. When the speed becomes normal, the upper pawl moves the lever H back into the lowest position, and the throttle valve is again opened. The *emergency gear* only comes into operation when the speed is abnormally increased, such as in cases of fracture of shaft, loss of the propeller, &c., in which cases the small weight A is left behind. This brings weight w into the position for closing the throttle valve, where it is held fast, and the valve remains closed until the weight is released by hand.

3. *Starting or bye-pass valves* are used to supply the M.P. or the L.P. valve chests or cylinders with live steam, in order that the engine may start away more easily. (See § 51.) If the cut-off in the cylinder is not less than 60 $^{\circ}/_{\circ}$, it is sufficient to introduce the steam into the valve chest. With earlier cut-offs it is better so to arrange the bye-pass valves, that steam can be admitted direct into the cylinder, on whichever side of the piston it may be required. In small engines, instead of an ordinary bye-pass valve, a dead-beat valve is often fitted, which is kept on its seat by the pressure of the steam, but can be raised at will by a lever. The diameter of the auxiliary steam pipes to each valve chest should be from $\frac{1}{3}$ to $\frac{1}{4}$ the diameter of the main steam pipe. If the steam is admitted direct to the cylinder, the diameter of each pipe may be rather less. This auxiliary steam should be taken from the main steam pipe to the engine, and controlled by a separate stop valve.

4. *Cylinder drain cocks* in vertical steam cylinders are only needed on the bottom, and should be so arranged that they can be easily worked by the engineer from the platform; and in larger engines they must therefore be fitted with the necessary gear for this purpose. Their diameter is from $\frac{1}{25}$ to $\frac{1}{50}$ the diameter of the cylinder; the lower value should be taken for large cylinders. Drain cocks of more than 2 inches in diameter are not usual, and instead of one large cock two smaller ones are then used, the discharges from which are connected to each other. The drain cocks should be placed below the cylinder bottom, and as low as possible in the passages. It is desirable, especially in the H.P. cylinder, to have a drain cock also on the cover side, because the water collecting at the top of the piston is only carried off very slowly with the exhaust steam. The drain pipes are generally led to the condenser, but they may also be carried to the hot well. In the latter case they must, however, be fitted with non-return valves, to prevent the water being sucked back. The drain cocks for the valve chests should be of the same size as those for the corresponding cylinders, and arranged in precisely the same manner.

5. *Relief valves* should be fitted both on the cover and on the bottom of the cylinder, to prevent the risk of fracture of the cylinder or rod, and allow water to escape, should it suddenly collect and "water-hammer" occur in the cylinder. Their diameters should be—

In the H.P. cylinder about $\frac{1}{12}$ to $\frac{1}{14}$ the diameter of the cylinder.
„ M.P. „ $\frac{1}{16}$ „ $\frac{1}{18}$ „ „
„ L.P. „ $\frac{1}{20}$ „ $\frac{1}{25}$ „ „

The relief valves should be surrounded with a casing, to prevent any one being scalded by the escaping water. In warships a small copper pipe is frequently led from this casing to the bilge. The springs of the valve should be set to about 1·2 times the maximum pressure which can occur in the valve chest of the corresponding cylinder. Relief valves are also generally fitted to the valve casings, when the auxiliary steam is led direct to the valve casing of the M.P. or L.P. cylinder, and should be of the same diameter as the relief valve on the corresponding steam cylinder.

6. *Indicator connections* and fittings are now provided for all main engines, and in warships also for some of the auxiliary engines. The connections to even the smallest cylinder should not be less than $\frac{1}{2}$ inch, and the pipes leading to the indicator should be of the same size. In large engines the indicator connections should be of larger diameter, at least from $1\frac{1}{4}$ to 2 inches, and may then be gradually reduced to $\frac{3}{4}$ inch as they approach the indicator cock. As a rule there is only one indicator cock to each cylinder. The two sides of the piston can be put into communication with the indicator cock, by means of the connecting pipes and a three-way cock. The latter should be either in the centre of the cylinder or at the upper end; the second arrangement is the better, because the top end is more likely to be free from water. The indicator drum should, if possible, be worked off an air-pump or similar lever, and, failing this, through a light lever from the crosshead. The gear should be as simple as possible. (See Part VI.)

7. If *jacket-heating* is required, the steam is led into the upper part of the cylinder jacket, and the condensed water drawn off from the lowest part, and led either direct or through a steam trap to the condenser. The diameter of the steam supply pipe to each cylinder should be $\frac{1}{2}$ to $\frac{3}{4}$ inch and of the drain pipe to the condenser $\frac{3}{8}$ to $\frac{5}{8}$ inch. A reducing valve is often fitted to each cylinder, and set to reduce the pressure in the jacket to that obtaining in the cylinder. It is advisable to have a small relief valve on each cylinder jacket.

§ 68. **Description of Figures 135 to 143** (drawings of actual

cylinders).—Fig. 135 shows the arrangement of cylinders for a compound engine of about 200 I.H.P., working with an initial pressure of $p = 100$ lb. per square inch. As usual in small engines, the cylinders have no liners. The H.P. cylinder has expansion gear and Rider valves, and the L.P. cylinder has flat D slide valves, with single ports. The H.P.

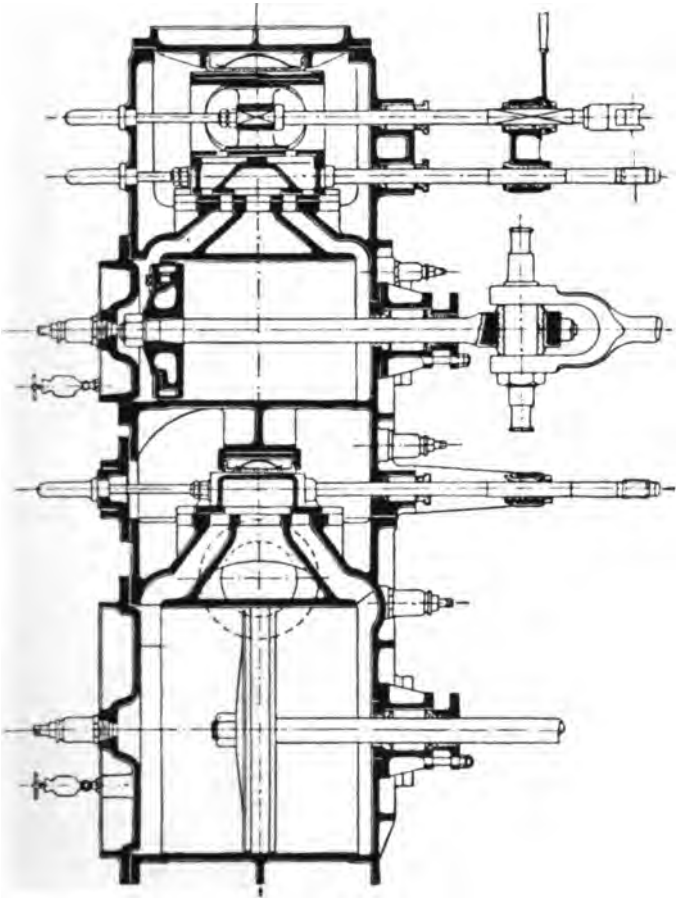


Fig. 135.

cylinder is fitted with a hardened valve face. The two cylinders are bolted together by means of a steam-tight joint. The slide valve of the L.P. cylinder can be taken out through the cover placed above it. The backs of the cylinders rest upon upright cast-iron frames and the fronts upon steel columns. (Compare Figs. 77, 78.)

Figs. 136 and 137 show the second M.P. cylinder, 2,640 mm. (8 feet

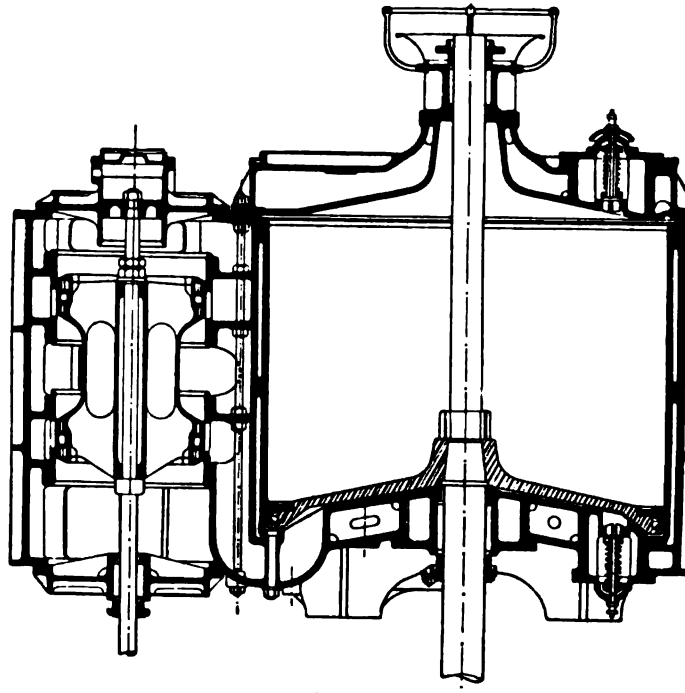


Fig. 136.

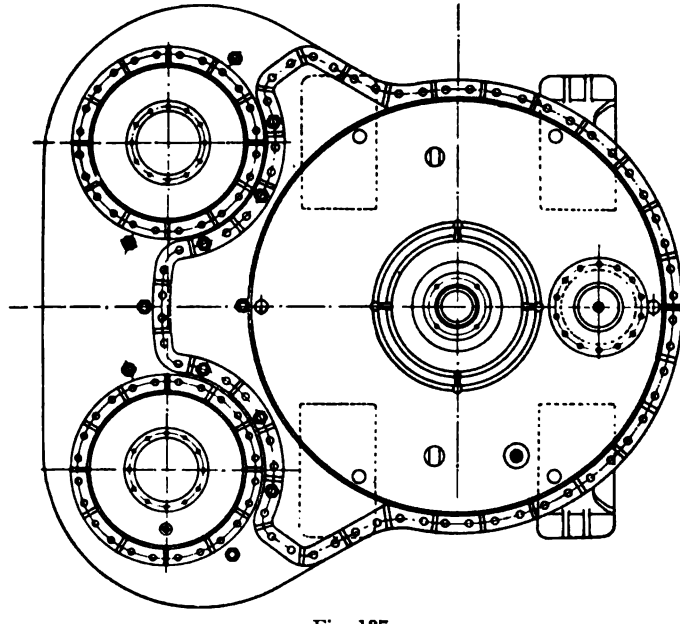


Fig. 137.

8 inches) diameter, of the quadruple engines of the ss. "Deutschland." The cylinder has a liner, two piston valves with steam on the outside, separate valve liners, and auxiliary steam cylinders to take the weight of the valves. The flat walls of the passages are strengthened with screwed

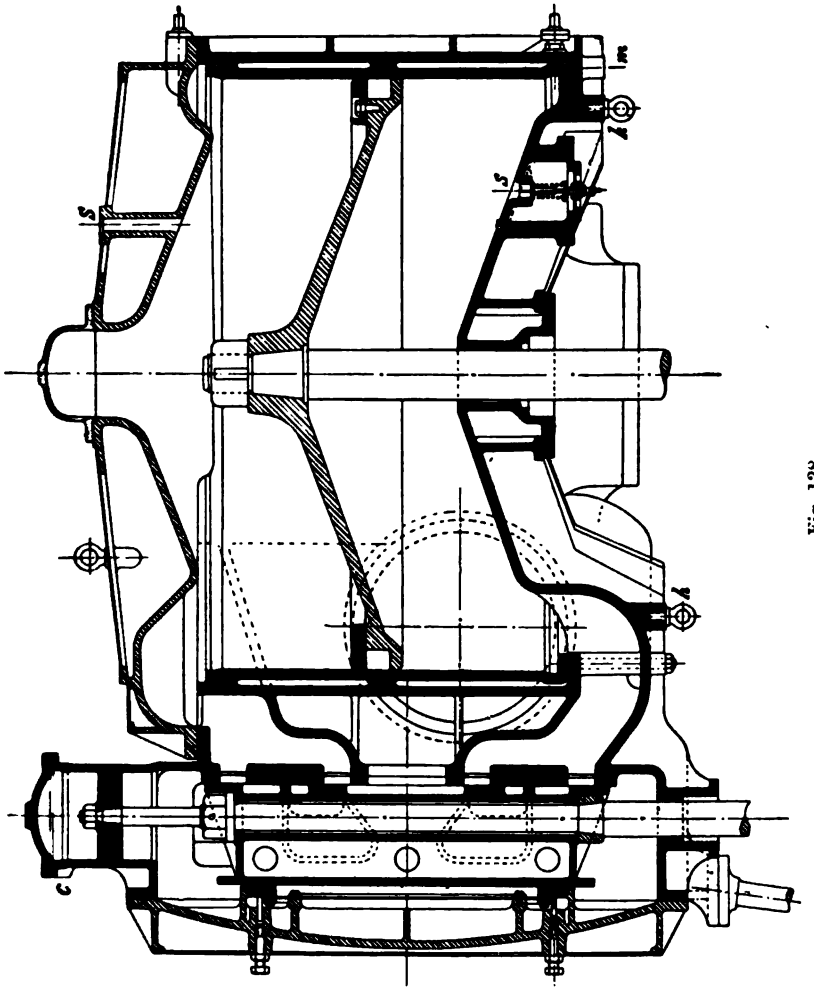


Fig. 138.

stays. The valves can be removed from above. The cylinder is cast with four feet, which are bolted to the cast-steel framing. Both the bottom and cover of the cylinder are made double, and each has a manhole and relief valve.

Figs. 138 to 141 show the L.P. cylinder, about 1,700 mm. (5 feet 7

inches) diameter, of an armoured cruiser. The cylinder has a liner, but the bottom and cover are single, the latter of cast steel. The stuffing-box is arranged for Philadelphia packing. The bottom is strengthened by radial ribs, and the ports by stay bolts, which also serve

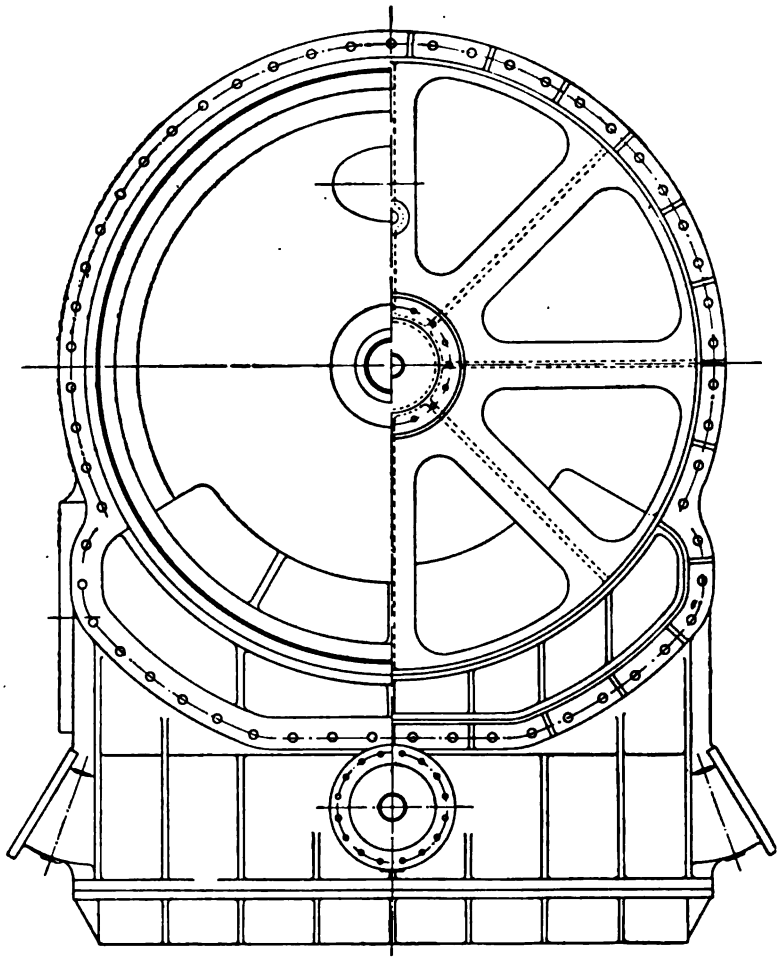


Fig. 139.

to secure the liner *a*. The cylinder rests on two cast-steel frames, to which it is bolted by two ribbed feet cast on to it. The weight of the overhanging valve chest is taken by two wrought-iron columns. On the cylinder bottom there are bosses, *h* for lifting eyes, *e* for draining the cylinder, *m* for draining the jacket. Relief valves *s* are

fixed to the cylinder cover and cylinder bottom. The valve is a double-ported slide valve, which works on an independent valve face bolted to the cylinder. The weight of the valve and valve rod is taken by a piston working in an auxiliary cylinder. The space above this balanced piston *c* is connected to the condenser. The pressure and therefore the friction of the valve on the face of the cylinder is relieved by a cast-iron ring of rectangular section placed at the back of the valve and connected

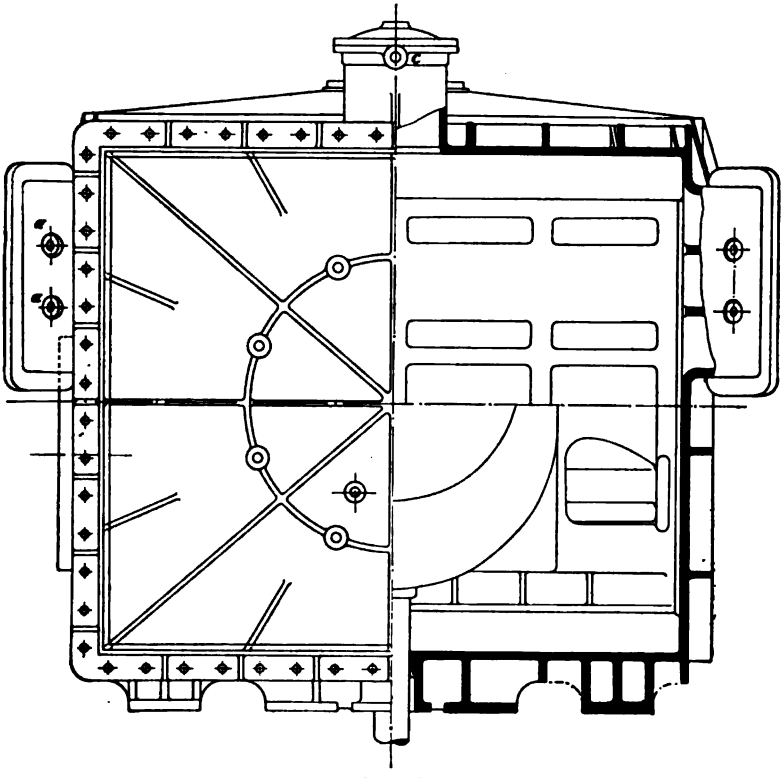


Fig. 140.

to the valve-chest cover by means of a copper diaphragm, which forms a flexible steam-tight joint. The ring is kept up against the back of the valve by springs. The space between the valve and the valve-chest cover, which is enclosed by the diaphragm, is connected to the condenser. The valve chest is of cast iron, and the cover of cast steel. Steam is admitted on both sides of the valve chest, but there is only one exhaust pipe.

Fig. 142, Plate XII., shows the arrangement of cylinders in a *destroyer* with triple-expansion engines and two L.P. cylinders. To

economise weight, none of the cylinders have liners, and all are made with single covers, top and bottom. The covers are of bronze. The H.P. cylinder has a centrally fed piston valve without a separate liner. The exhaust from the H.P. is led, by means of a pipe running

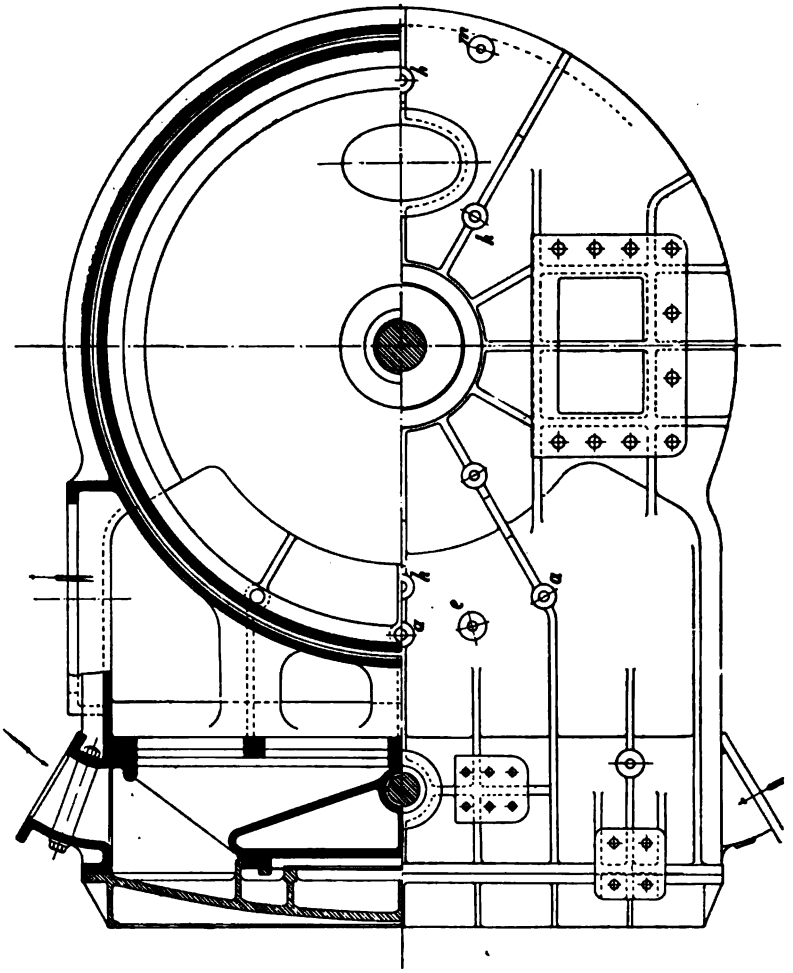
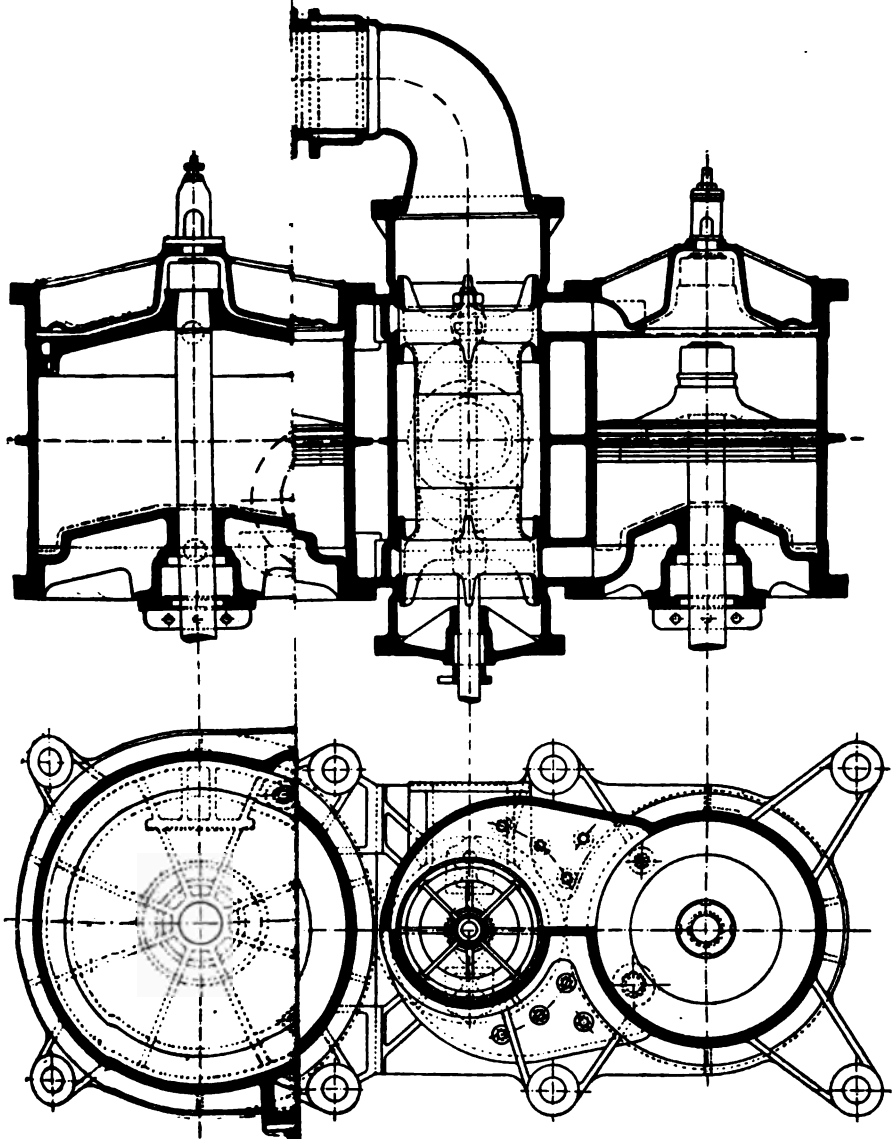


Fig. 141.

over the M.P. cylinder, to the M.P. valve casing. The M.P. piston valve takes steam at both ends, has a special liner and piston rings. The L.P. cylinders have double-ported unbalanced flat slide valves, working direct on the valve faces, which are cast in one with the cylinders. Each L.P.

PLATE XII.



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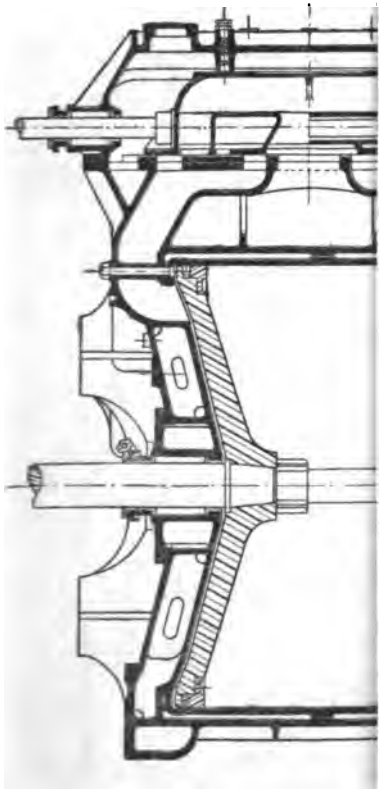

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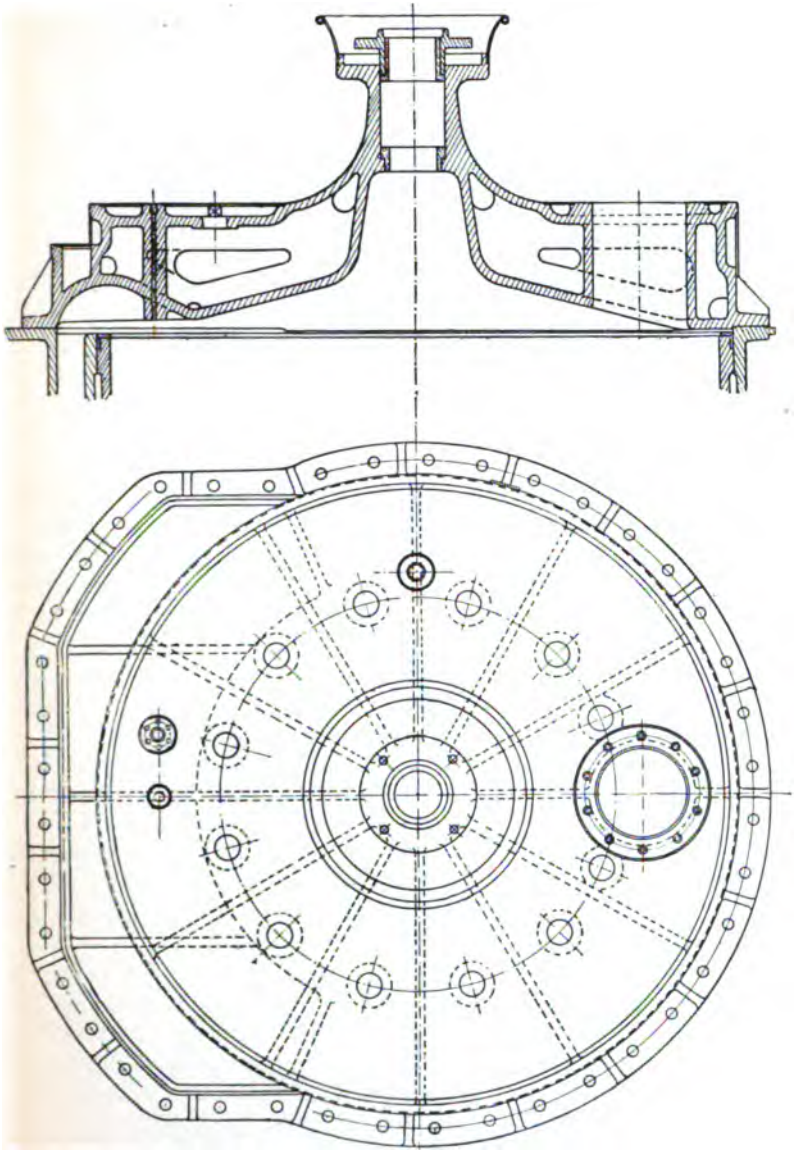


Fig. 144.

cylinder exhausts through a separate exhaust pipe at the side. All the piston-rod stuffing-boxes are arranged for United States packing. The cylinders are supported by steel columns. The engine is balanced on

the Schlick system, hence the thick pistons in the M.P. and first L.P. cylinders. The crosshead guides are bolted to the columns and to lugs cast on the cylinder bottoms.

Fig. 143, Plate XIII., shows the L.P. cylinder of the twin-screw steamer "Deutschland," on the cover of which the H.P. cylinder is fitted. The L.P. cylinder is entirely jacketed both at the sides and on the top and bottom. There are two double-ported slide valves side by side, each having an auxiliary steam cylinder and a balanced ring at the back of the valve. The H.P. cylinder is secured to the cover of the L.P. cylinder by a casting and two strong cast-iron feet. The H.P. cylinder also is entirely jacketed, and is fitted with a piston valve working in hard fine-grained cast-iron liners forced into the valve chest. The two stuffing-boxes between the cylinders, as well as the stuffing-box of the L.P. cylinder, are fitted with metallic packing. The flat surfaces of both cylinders are stayed.

§ 69. **Cylinder Covers.**—These are made either single or double. Single covers are used either for the sake of cheapness and simplicity, or, as in warships, to effect a saving in weight. *Double cylinder covers* are employed in large merchant vessels, where it is intended to steam-jacket the covers as well as the cylinder walls. They generally have radial ribs in which openings are provided to allow the water to drain away, as well as to connect the cores. (See Figs. 144, 156.) In *single cylinder covers* the ribs often have a T-shaped section. It is very important that the cylindrical or cup-shaped recess, provided in the centre of the cover for the piston-rod nut, should be very solid. (See Fig. 145.) The inner surface of the cylinder cover must be made as far as possible of the same shape as the piston, leaving a uniform clearance of from $\frac{1}{8}$ to $\frac{3}{4}$ inch, according to the size of the cylinder. This is to reduce the clearance to a minimum, but as irregularities in the castings cannot be wholly avoided, less space than that mentioned above should not be allowed.

Cylinder covers are usually made of cast iron in the mercantile marine, of cast steel in warships, and in the case of very small and light engines generally of gunmetal. As the strain on the cylinder cover can only be calculated approximately, and its shape is usually complicated, the thickness a must be largely determined by actual experience. For cast iron this thickness is—

1. For flat single covers a = from 0·9 to 0·95δ
2. For conical single covers a = „ 0·8 „ 0·85δ
3. For double covers a = „ 0·75 „ 0·85δ

δ being the thickness of the cast-iron cylinder lining. (See Table No. 17, page 130.)

Cylinder covers of cast steel are generally made single, and about 60 to 65 % the thickness of a similar cast-iron cover.

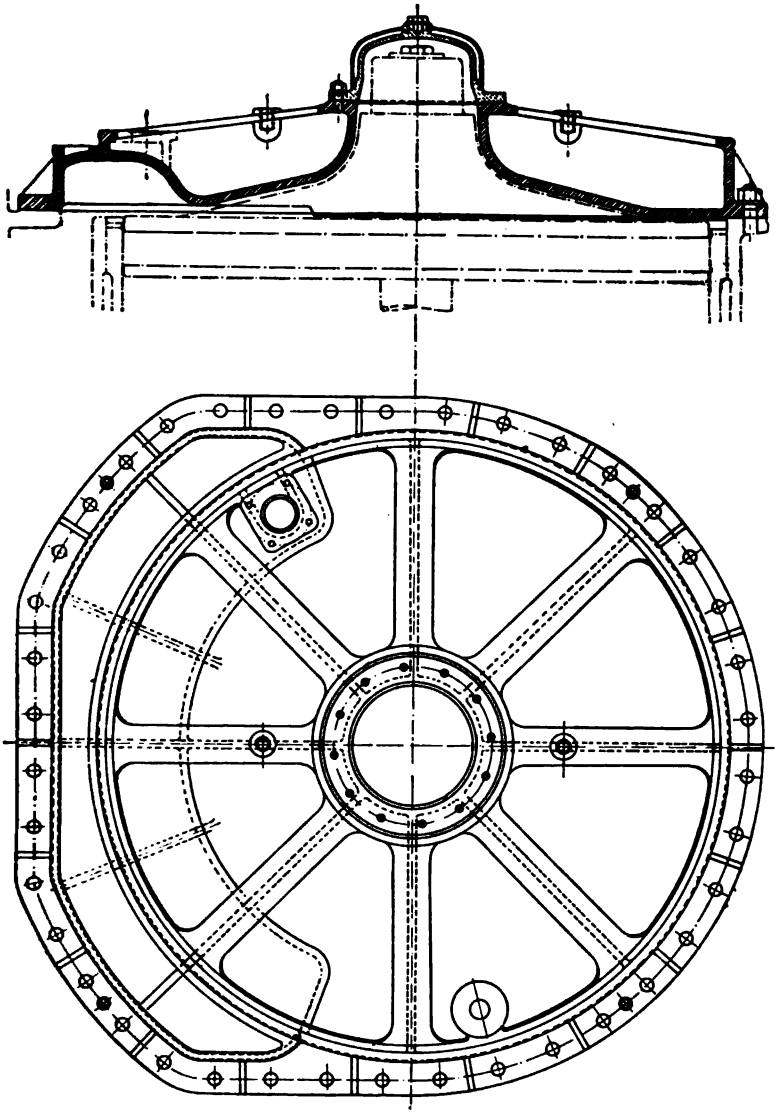


Fig. 145.

The height h of the radial ribs as well as the clear space between the two walls of a double cylinder cover is generally about $h = 7a$ to

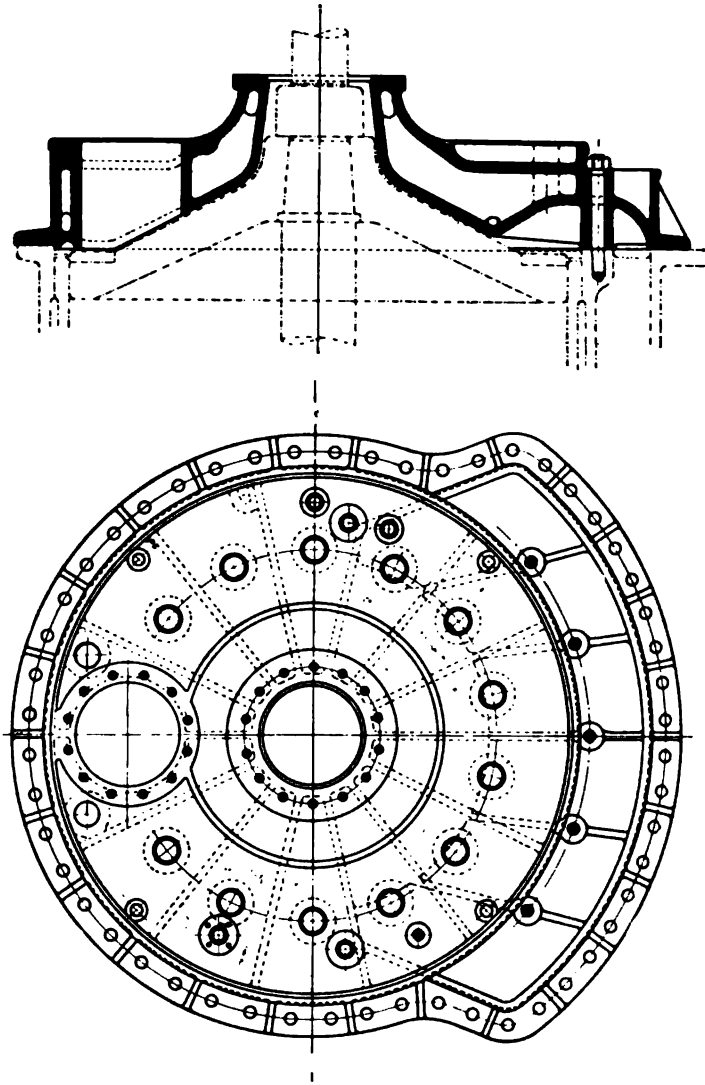


Fig. 146.

$9a$; thickness of the ribs b = about $0{\cdot}9\delta$. The thickness and width of the flange of the cylinder cover depend on the diameter d_2 of the cover studs or bolts, as follows:—

Thickness of the flange t = from $1{\cdot}15$ to $1{\cdot}25d_2$
Width „ „ = „ $2{\cdot}6$ „ $3{\cdot}3d_2$

For diameter of cylinder cover studs or bolts see § 58, page 133.

If the piston rod passes through the top cover so as to form a guide to the piston, it is either made steam-tight by a stuffing-box, or it runs in a closed cast-iron casing bushed at the bottom with gunmetal. A lubricator should be fitted at the top or side of the casing.

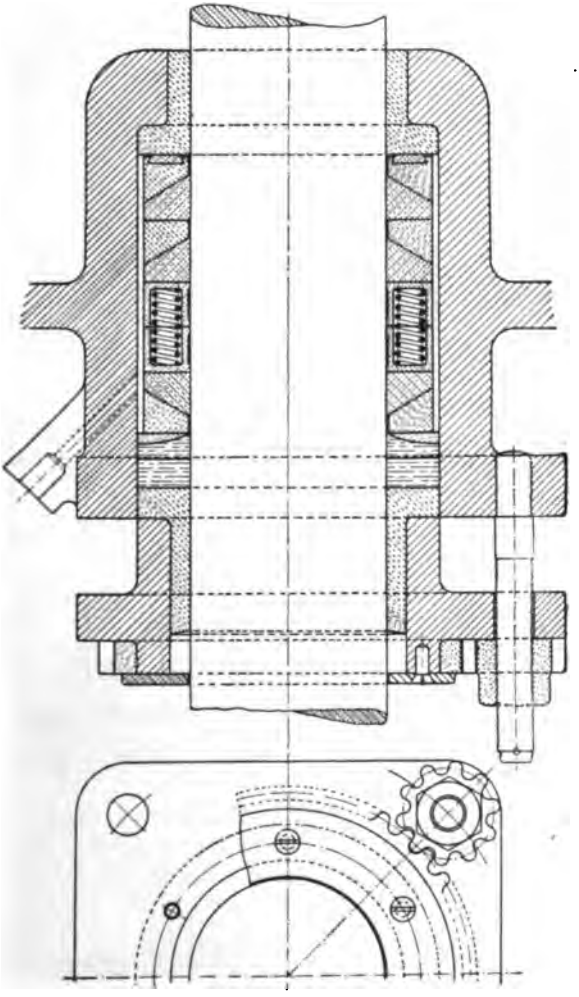


Fig. 147.

Large cylinder covers of above 4 feet diameter are generally provided with a manhole and cover, so that the piston or cylinder can be examined, without having to lift the large cylinder cover. (See Figs.

143, 144, and 146.) The manhole should be about 15 to 16 inches in diameter, or, if elliptical, 11 by 15 inches. In order to take off the cylinder or manhole cover easily, from two to five tapped holes should be provided, into which strong tap-bolts can be fitted, for prizing up the covers. Two strong eye-bolts on the covers, exactly opposite each other, should also be provided, to which the lifting gear for raising the covers can be attached.

General Remarks.—The covers of all cylinders are lagged with a non-conducting composition, over which is placed a covering of sheet iron or of chequer plate, if the cover be large and much trodden on. If the covers are steam-jacketed the core-holes must be carefully closed.

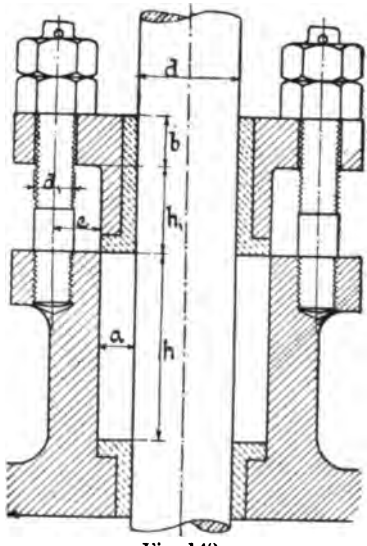


Fig. 148.

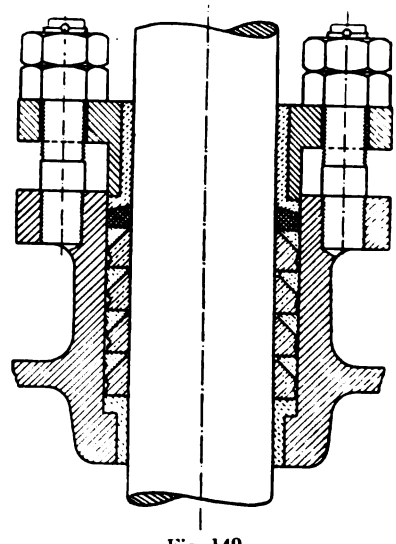


Fig. 149.

This is generally done with slightly conical screw plugs with a fine thread, firmly screwed home. If the holes are merely rimered out and plain taper plugs simply driven in, the covers are liable not to be quite so steam-tight.

§ 70. **Stuffing-boxes.**—Soft packing is generally used—for steam, asbestos or "Tucks" packing, with interwoven wire; and for water, greased hemp rope. The depth and thickness of the packing are determined by the pressure to which the stuffing-box is exposed. Table No. 18 gives the usual dimensions of these stuffing-boxes for pressures of 140 to 170 lb. per square inch. For higher pressures the depth of the packing space must be increased. In stuffing-boxes for rods up to $1\frac{1}{2}$

TABLE No. 18.

Dimensions of Stuffing-boxes for Ordinary Packing (Fig. 148).

d Diameter of Rod.			a Thickness of Packing.			h Depth of Packing Space.			h_1 Length of Stuffing-box.			n Number of Screws	d_1 Diameter of Screws, Inches.
Mm.	Inches.		Mm.	Inches.		Mm.	Inches.		Mm.	Inches.			
		Approx.			Approx.			Approx.			Approx.		
15	0·59	$\frac{5}{8}$	8	0·31	$\frac{5}{16}$	40	1·58	$1\frac{9}{16}$	15	0·59	$\frac{5}{8}$	2	$\frac{1}{2}$
20	0·79	$\frac{3}{4}$	10	0·39	$\frac{3}{8}$	50	1·97	$1\frac{15}{16}$	18	0·71	$\frac{11}{16}$	2	$\frac{1}{2}$
25	0·98	1	11	0·43	$\frac{7}{16}$	60	2·36	$2\frac{3}{8}$	22	0·87	$\frac{7}{8}$	2	$\frac{1}{2}$
30	1·18	$1\frac{3}{16}$	12	0·47	$\frac{1}{2}$	70	2·76	$2\frac{3}{4}$	25	0·98	1	2	$\frac{1}{2}$
35	1·38	$1\frac{3}{8}$	13	0·51	$\frac{1}{2}$	80	3·15	$3\frac{1}{8}$	30	1·18	$1\frac{3}{16}$	2	$\frac{1}{2}$
40	1·58	$1\frac{9}{16}$	14	0·55	$\frac{9}{16}$	85	3·35	$3\frac{5}{16}$	35	1·38	$1\frac{3}{8}$	2	$\frac{1}{2}$
50	1·97	2	16	0·63	$\frac{5}{8}$	95	3·74	$3\frac{3}{4}$	40	1·58	$1\frac{9}{16}$	2	$\frac{1}{2}$
60	2·36	$2\frac{3}{8}$	16	0·63	$\frac{5}{8}$	105	4·13	$4\frac{1}{8}$	45	1·77	$1\frac{3}{4}$	2	$\frac{1}{2}$
70	2·76	$2\frac{3}{4}$	18	0·71	$\frac{11}{16}$	120	4·72	$4\frac{3}{4}$	55	2·17	$2\frac{1}{8}$	2	$\frac{1}{2}$
80	3·15	$3\frac{1}{8}$	18	0·71	$\frac{11}{16}$	135	5·31	$5\frac{5}{16}$	60	2·36	$2\frac{3}{8}$	2	$\frac{1}{2}$
90	3·54	$3\frac{9}{16}$	20	0·79	$\frac{3}{4}$	150	5·90	$5\frac{13}{16}$	70	2·76	$2\frac{3}{4}$	3	$\frac{1}{2}$
100	3·94	$3\frac{15}{16}$	20	0·79	$\frac{3}{4}$	160	6·29	$6\frac{5}{16}$	70	2·76	$2\frac{3}{4}$	3	1
110	4·32	$4\frac{5}{16}$	22	0·87	$\frac{7}{8}$	170	6·69	$6\frac{11}{16}$	80	3·15	$3\frac{1}{8}$	3	1
120	4·72	$4\frac{3}{4}$	22	0·87	$\frac{7}{8}$	180	7·08	$7\frac{1}{16}$	80	3·15	$3\frac{1}{8}$	3	1
130	5·11	$5\frac{1}{8}$	25	0·98	1	190	7·47	$7\frac{1}{2}$	90	3·54	$3\frac{9}{16}$	3	$1\frac{1}{8}$
140	5·51	$5\frac{1}{2}$	26	1·02	1	200	7·87	$7\frac{7}{8}$	90	3·54	$3\frac{9}{16}$	4	$1\frac{1}{8}$
150	5·90	$5\frac{13}{16}$	28	1·10	$1\frac{1}{8}$	210	8·26	$8\frac{1}{4}$	100	3·94	$3\frac{15}{16}$	4	$1\frac{1}{8}$
160	6·29	$6\frac{5}{16}$	30	1·18	$1\frac{3}{16}$	220	8·66	$8\frac{11}{16}$	100	3·94	$3\frac{15}{16}$	4	$1\frac{1}{8}$
170	6·69	$6\frac{11}{16}$	32	1·25	$1\frac{1}{4}$	230	9·05	$9\frac{1}{16}$	100	3·94	$3\frac{15}{16}$	4	$1\frac{1}{4}$
180	7·08	$7\frac{1}{16}$	32	1·25	$1\frac{1}{4}$	245	9·64	$9\frac{5}{8}$	110	4·32	$4\frac{5}{16}$	4	$1\frac{1}{4}$
190	7·47	$7\frac{1}{2}$	35	1·38	$1\frac{3}{8}$	260	10·23	$10\frac{1}{4}$	110	4·32	$4\frac{5}{16}$	4	$1\frac{1}{4}$
200	7·87	$7\frac{7}{8}$	35	1·38	$1\frac{3}{8}$	275	10·82	$10\frac{13}{16}$	110	4·32	$4\frac{5}{16}$	4	$1\frac{1}{4}$

inch diameter the gland is generally of gunmetal; but for rods of larger diameter the gland is made of cast iron bushed with gunmetal. The clearance between the neck bush and rod, and between the gland and rod is about $\frac{1}{240}$ the diameter of the rod. The thickness of the flange of the stuffing-box collar is $b = 1{\cdot}25$ to $1{\cdot}3d_1$, d_1 being the diameter of the gland bolts. The distance of the latter from the centre of the rod should be as small as possible, in order to avoid bending the flange of the gland. The distance c (Fig. 148) from the centre of the bolt to the outer face of the gland is generally about $1{\cdot}0d_1$ to $1{\cdot}25d_1$.

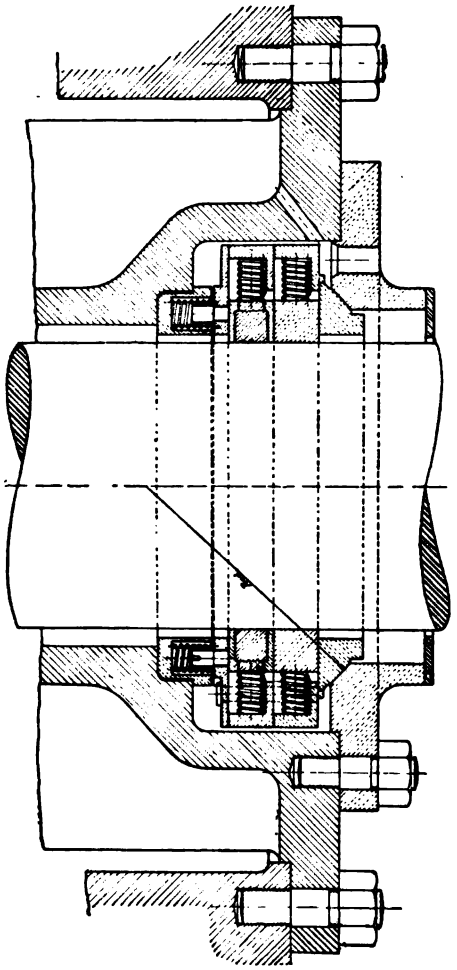


Fig. 150.

In order that the nuts of the stuffing-box may be turned equally they are frequently shaped like cog wheels, and engage in a common circular rack, thus enabling them to be turned at the same time. (See Fig. 147.)

§ 71. **Metallic Packings.**—In modern practice stuffing-boxes are frequently packed with *metallic packings*, of which there are a large number of different kinds. Fig. 149 shows the "*Katzenstein packing*," which consists of a number of conical rings divided into two or three parts. The rings next the piston rod are of white metal, those between them of bronze. In order that each ring may exert an elastic pressure, one or two turns of common hemp packing are laid over the top ring, and held in place by the gland. This packing does not allow of any lateral movement of the rod. In the

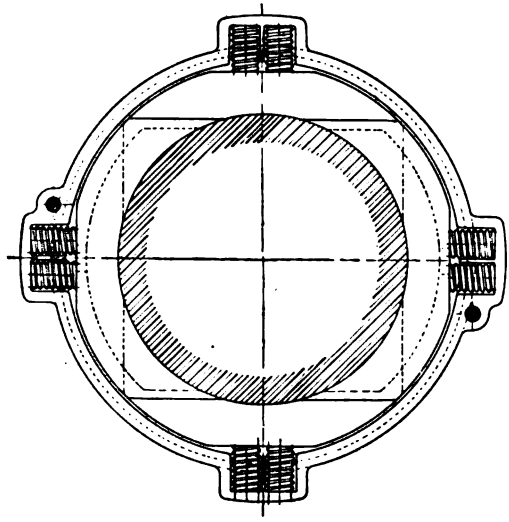


Fig. 151.

"*Schelling packing*," shown in Fig. 147, the outer metallic rings are in one piece, the inner, of white metal, in three or four parts. The bottom of the packing space and the uppermost metallic ring are turned smooth. A slight lateral motion is allowed to the piston rod, as the space round the outer rings gives a small amount of play. This space is connected through a small cock with the condenser, to draw off condensed steam. Two pieces of soft rope are laid over the metal rings, and pressed down by the gland.

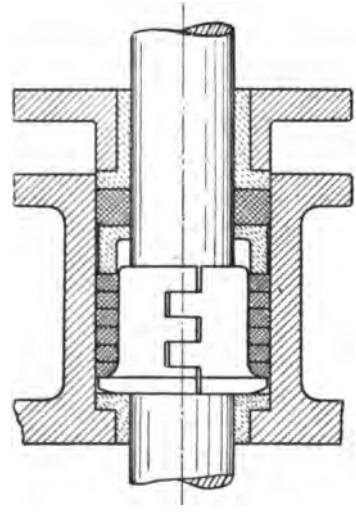


Fig. 151A.

Another kind of "*Schelling packing*" is shown at Fig. 151A. The rod is enclosed within a ring made in two halves and dovetailing into one another. The ring is composed of an alloy of copper which fuses at a red heat. Soft packing is fitted at the back of this ring. The ring and the packing are held in position axially by several other turns of soft packing and an adjustable gland, the object being to allow the rod sufficient lateral play.

Figs. 150 and 151 show the "*United States packing.*" It consists of two metallic rings, each in four parts, ground to fit the one upon the other. Two of the parts opposite each other are filled in with white metal, while the two other parts are solid, and fit into them exactly. The two rings are placed at an angle of 90° to one another. They are pressed against the rod by lateral springs contained in a casing enclosing the inner rings. At the bottom the rings bear upon another massive ring, which is made steam-tight on the side next the gland by means of a spherically shaped cover-piece. The rings are held together at the top by springs resting against the bottom of the stuffing-box. This packing also allows of slight lateral movements of the piston rod. The gland of the stuffing-box has a drain cock, to carry off the condensation water to the condenser. Of late what is known as *electro-deposited metallic paper packing* has been used for superheated steam. (See Fig. 152.) It consists of a large number of rings made of a very soft electrically deposited metallic preparation, packed one above the other. A piece is cut out of the rings, so that when they are fitted into the packing space they assume a conical shape. To keep the lubricant in the stuffing-box, one or two turns of soft packing are laid on the top of the metallic rings.

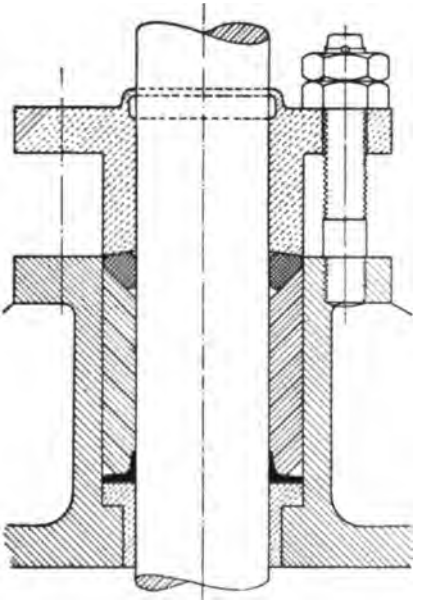


Fig. 152.

Valves.

§ 72. **General Remarks.**—Slide valves are now almost exclusively used in marine engines. Unbalanced flat slide valves are generally employed when the steam pressure does not exceed 100 lb., but piston valves are, as a rule, used for higher pressures. As ordinary flat slide valves require a considerable amount of power to drive them, they are frequently fitted with an arrangement designed to relieve the pressure of the slide valve on the face of the cylinder (Fig. 158). The space between the valve and the cover, shut off by the packing ring, is, as a rule, connected to the condenser. (See Cylinder, Figs. 138 and 143.) The desire to relieve the pressure on the back of the slide valve has led to the adoption of the so-called covered valves (Fig. 153). The valve, the two slide faces of which are exactly alike, is made to slide with as little play as possible between the actual valve face and the cover fitted over it, the working surface of which is exactly the same as that of the valve face. As may be seen from the figure, the valve has double steam and exhaust ports. The valve cover has two sides resting upon the valve face, and it must be sufficiently strong not to bend or sag from the pressure of the steam on the back. A strong spring holds it down, and two lugs prevent any movement longitudinally. The usual types of slide valves are shown in Figs. 154 to 159.

When flat slide valves are used care must be taken that the valve and valve face do not bend under the pressure of the steam, and that the bearing surfaces are large enough to take up the pressure of the valve without undue friction. To relieve the valve to some extent, and lubricate it at the same time, the valve faces or the slide valves frequently have grooves cut in them, from $\frac{3}{8}$ to $\frac{5}{8}$ inch wide, and from $\frac{1}{16}$ to $\frac{1}{8}$ inch deep, or smooth flat-bottomed holes drilled in them from $\frac{5}{8}$ to $\frac{3}{4}$ inch in diameter, to allow freer access of the steam between the working surfaces. Piston valves, owing to their shape, need no balancing, but they have the disadvantage of not being as steam-tight as slide valves. In piston valves fitted with spring rings, or with rings held out by means of springs, the valve liners are apt to wear unevenly. It has, therefore, been the practice of late to make piston valves with solid rings. These, though not so steam-tight as spring rings, have been found to wear well. The rings are often cut, and the ends firmly fastened

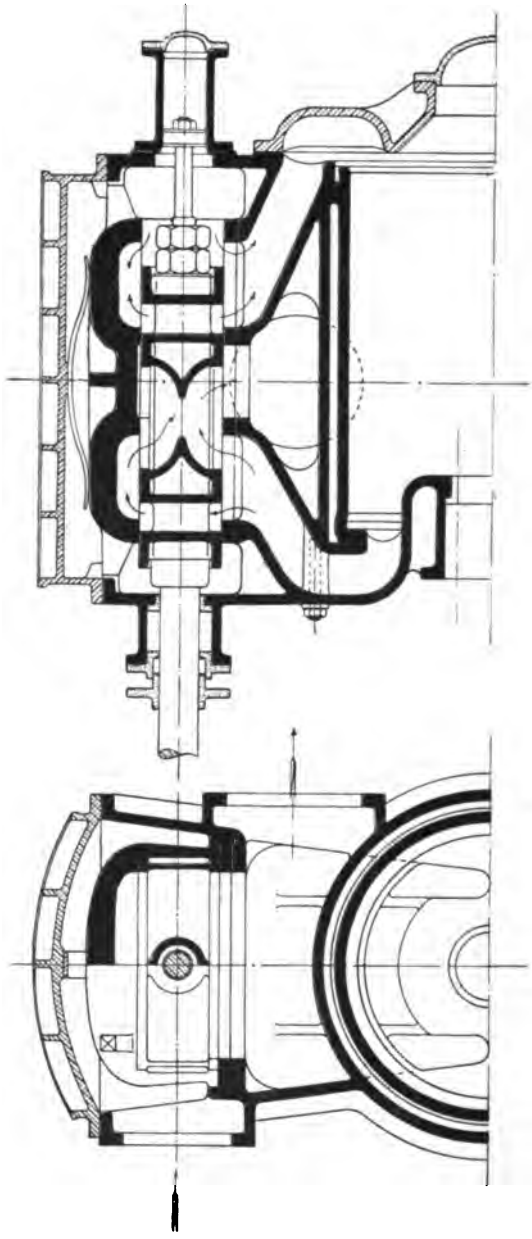


Fig. 153.

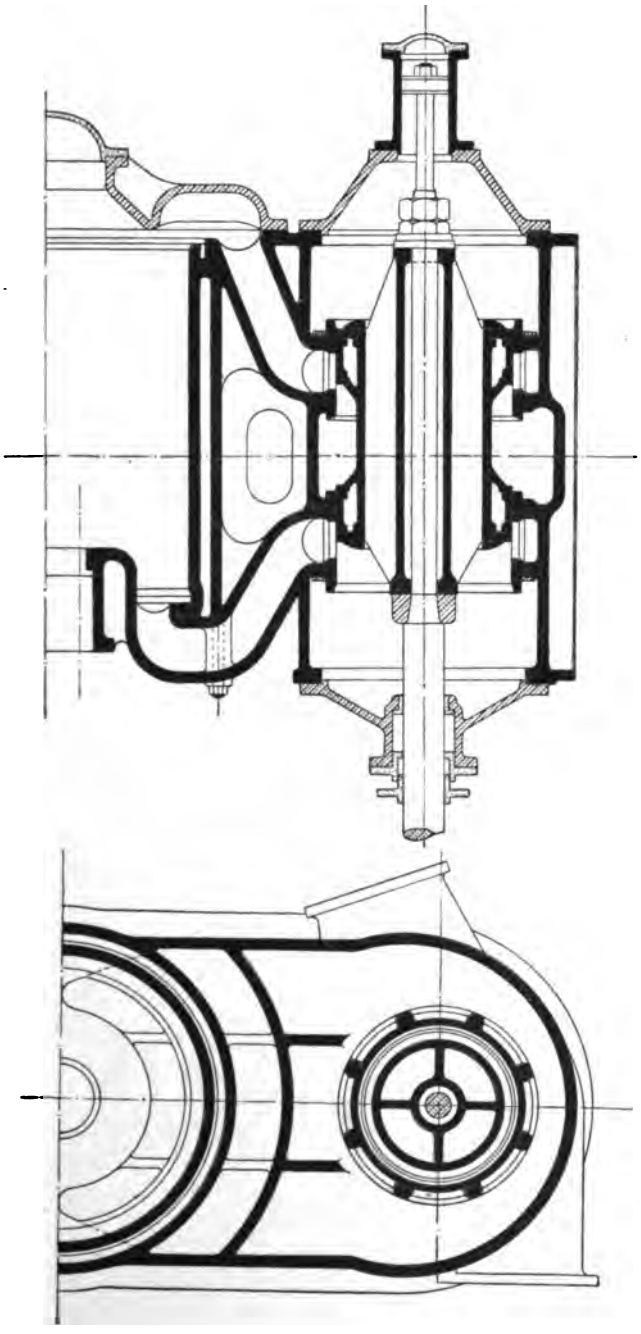


Fig. 154.

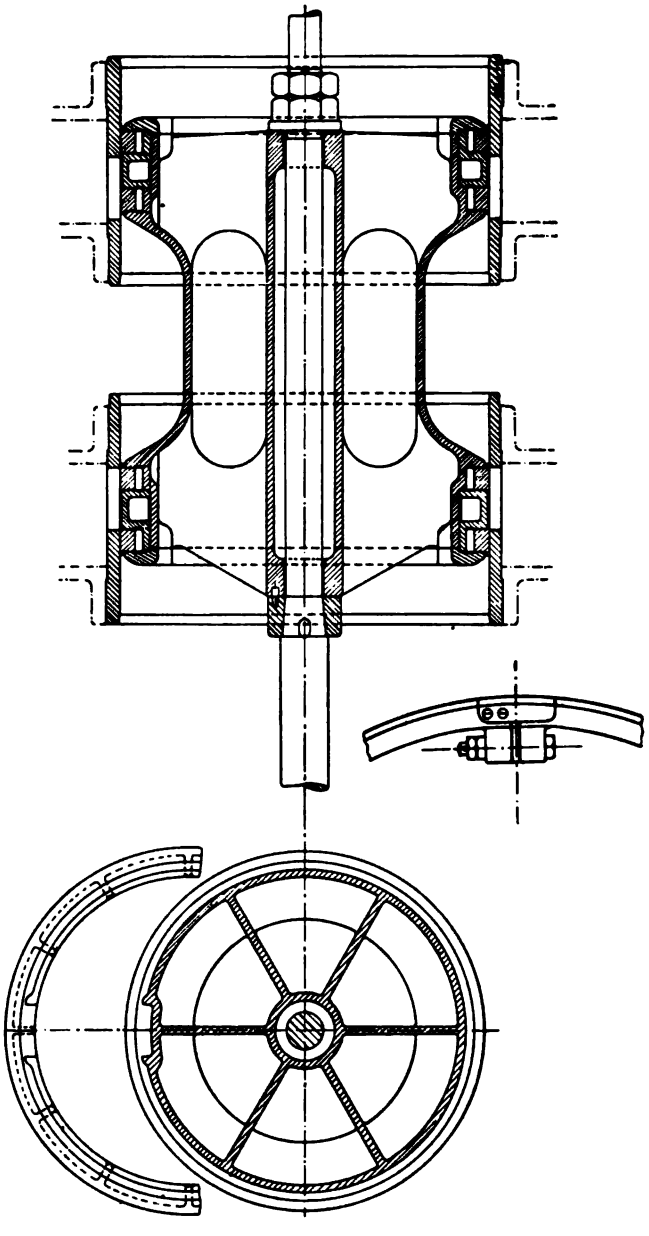


Fig. 155.

with screws. This enables the diameter of the rings to be increased by inserting thin strips of sheet metal as may be required to take up the wear. (See Fig. 155.) The adjusting screws are often so arranged that the edges of the rings can be brought together, and the opening entirely closed. To prevent the rings being forced too far apart, and thus becoming jammed in the cylinder, lock nuts are provided.

The material both of the valve and of the valve face is generally cast iron. When flat slide valves are used the valve faces are frequently separate, and bolted to the cylinder; they are held in place by set screws of steel, wrought iron, or "Delta"* metal, the heads being countersunk. (See Fig. 127.) Piston valves generally work in separate cast-iron liners, in which the ports are cut. (See Fig. 126.) The liner is occasionally simply pressed into the cylinder or valve-chest casting, but it may also be fixed in the way shown at Fig. 126.

§ 73. **The Thickness (s_1) of Piston Valve Liners** is about $= \frac{D}{30} + \cdot 39$, D being the internal diameter of the valve liner. The webs of the valve, which are generally at an angle of 60° to each other (see Fig. 126), should have a thickness $s =$ from 1 to $1 \cdot 2 s_1$.

§ 74. **Ports of Valve Face.**—The ports of the valve face are usually as long as circumstances will allow, and their height correspondingly small. The length of the ports is as a rule from 0·9 to 0·95 the diameter of the cylinder. The effective cross-sectional area of the openings in the valve face is made about 5 to 10 % greater than the cross-sectional area of the port, especially in double-ported valves. In piston valves the effective area of the openings in the valve face are generally about 10 to 20 % more than the area of the port, as the various webs across the openings offer more resistance to the steam than do the smooth walls of the port. The effective circumferential area of the valve face is about 65 to 75 % of the inner circumferential area of the valve liner. The exhaust port should be made so large that, even when the valve is at the dead point, the actual area of the opening is not less than the cross section of the port.

§ 75. **Symbols used in connection with Slide Valves.**—In the following paragraphs—

r = amount of eccentricity = half travel of eccentric.
δ = angle of lead.
a = height of the port in the valve face.
e = outside lap.
i = inside lap. (See Fig. 156, &c.)

* Bronze screws, of whatever kind, offer little resistance to torsional stress, and break off easily, if the strain on them, when putting them in, is too great.

§ 76. **Stroke of Valve.**—As the friction of the valve varies approximately with its stroke, the latter is made as short as practical conditions will allow. To diminish the travel of the valve, the larger cylinders have double or treble ported valves (Figs. 158 and 159). The latter, however, are only used under exceptional conditions, as they require too great a length of valve and valve chest. To get a quick admission and a large opening for steam, "Trick" valves (Fig. 157) are often used instead of the ordinary D slide valves. The steam is exhausted in the same way as in the ordinary D slide valves.

§ 77. **The Amount of Eccentricity,** or half the stroke of the valve, is usually r = from 1 to 1·4a. It is generally so chosen that the largest steam port opening (mean value of top and bottom) is as follows:—

In the H.P. slide valve about 0·8
" M.P. " " 0·75 } of the area of the port.
" L.P. " " 0·7

In piston valves the largest mean steam port opening is generally made somewhat greater than that given above, on account of the various webs.

§ 78. **Principal Dimensions.**—1. *Ordinary* D *slide valve* (Fig. 156)—

Thickness of the walls of the valve δ_1 = from 0·5 to 0·6 the thickness of the cylinder walls.

Thickness of the slide face f = from 0·8 to 0·9 the thickness of the cylinder walls.

2. *Trick valve* (Fig. 157)—

$a_1 \leqq a + b$ (a being = height or width of the port for an ordinary D slide valve). (See Fig. 156.)

$a_2 = \frac{a}{4}$ to $\frac{a}{2}$. $b = \frac{3}{8}$ to $\frac{3}{4}$ inch. $c_o = e_u$. $c_u = e_o$.

δ_1 = 0·5 to 0·6 thickness of cylinder wall.
f = 0·8 to 0·9 " "

3. *Double-ported slide valve* (Figs. 158 and 159)—

$a_1 = r - e$. $b = \frac{3}{4}$ to $1\frac{5}{8}$ inch. $d = r - i$. $c = 2r + b$.
δ_1 = 0·5 to 0·6 thickness of cylinder wall.
f = 0·8 to 0·9 " "
δ_2 = 1·0 to 1·1 " "

§ 79. **The Cut-off** can be taken at the following values for valves having a full stroke or travel:—

In the H.P. cylinder about 60 to 75 %.
„ M.P. „ „ 55 „ 70 %.
„ L.P. „ „ 50 „ 65 %.

The work in the different cylinders can be equalised, as required, by adjusting the respective links and gear.

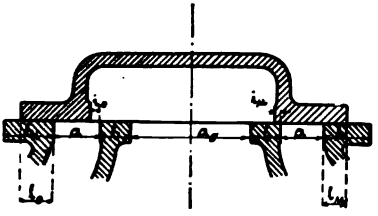


Fig. 156.

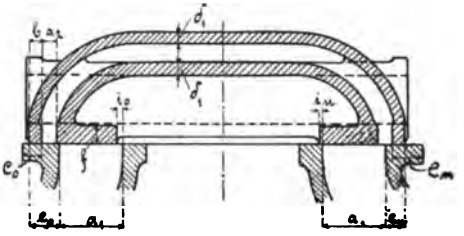
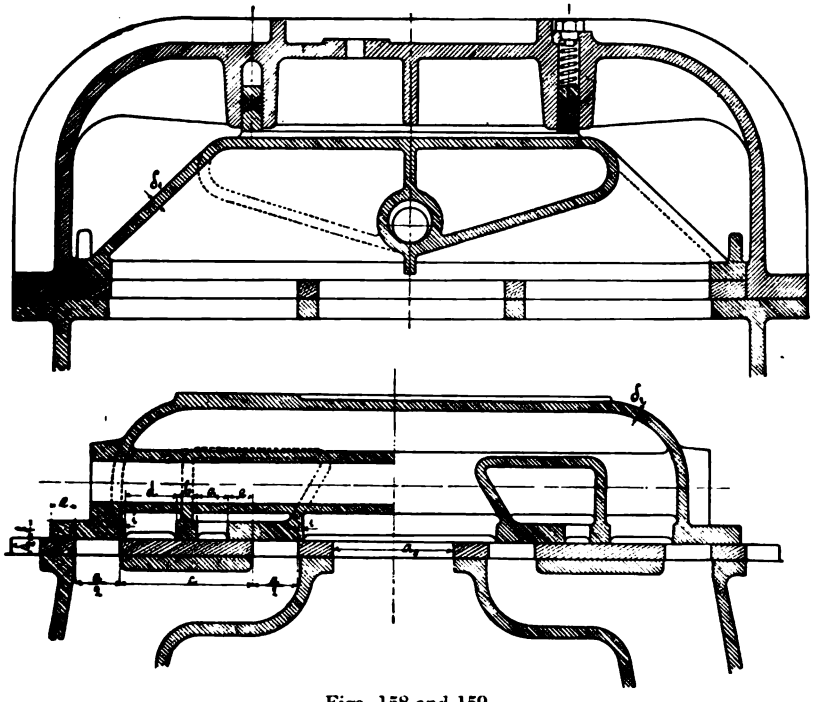


Fig. 157.



Figs. 158 and 159.

In compound engines, with cranks at 90°, it is advisable to have a maximum cut-off of at least 60 %, because the engine can then be started by admitting live steam to the receiver. If the cut-off is less,

it is necessary to admit this steam to the top and bottom sides of the low-pressure piston. (See § 51.)

§ 80. **Linear Lead.**—By this term is meant the port opening (in inches), when the piston is at the dead point; this varies from $\frac{1}{16}$ in small to $1\frac{3}{16}$ inch in large engines. The lead is generally twice as large at the bottom as at the top end of the cylinder, in order that the cut-off may be as uniform as possible at both ends. The length of the connecting rod would otherwise cause it to be from 1 to 2 % less at the bottom than at the top end. In a valve working at its full stroke, the following values may be taken for the lead:—

In the H.P. cylinder,	top	end,	$\frac{r}{14}$	to	$\frac{r}{6}$
„ „ „	bottom	„	$\frac{r}{7}$	to	$\frac{r}{4{\cdot}5}$
„ M.P. „	top	„	$\frac{r}{13}$	to	$\frac{r}{5{\cdot}3}$
„ „ „	bottom	„	$\frac{r}{6{\cdot}5}$	to	$\frac{r}{3{\cdot}9}$
„ L.P. „	top	„	$\frac{r}{12}$	to	$\frac{r}{4{\cdot}8}$
„ „ „	bottom	„	$\frac{r}{5{\cdot}8}$	to	$\frac{r}{3{\cdot}3}$

The following angles α between the position of the crank at which the admission port begins to open, and the dead point, correspond approximately to the linear leads given above:—

In the H.P. cylinder	4°	to	10°	at the top	end.
„ „ „	8°	„	13°	„ bottom	„
„ M.P. „	5°	„	11°	„ top	„
„ „ „	9°	„	15°	„ bottom	„
„ L.P. „	6°	„	12°	„ top	„
„ „ „	10°	„	18°	„ bottom	„

It should be noted that the lower values refer to slow-running engines with speeds of from 70 to 80 revolutions per minute, and the higher to those running at 350 revolutions per minute and upwards. For engines running at other speeds, values between those given above should be selected.

§ 81. **Exhaust Lead.**—This may be smaller with slow speed than with high speed engines, and is usually—

7 to 14 %	of the stroke	in the H.P.	cylinder.
9 „ 18 %	„	„ M.P.	„
11 „ 22 %	„	„ L.P.	„

It is also desirable that this lead should be 15 to 20 % more at the

bottom than at the top end of the cylinder, so that the steam, during the rapid change at the top of the stroke, can get away at the bottom quickly enough.

§ 82. **Compression.**—The degree of compression is so arranged that the final maximum pressure shall not be greater than the pressure in the valve chest. It varies from—

4 to 8 % of stroke in the H.P. cylinder.
7 „ 14 % „ „ M.P. „
10 „ 20 % „ „ L.P. „

It may be taken as somewhat higher in compound than in triple engines, because in the former the difference of pressure in each cylinder is greater. Like exhaust lead, or inside lap, compression should be greater at the bottom than at the top end of the cylinder. On each side of the piston the one is directly dependent on the other, a small exhaust lead or inside lap producing a high degree of compression. The two must therefore be made to correspond with each other, for any given working conditions.

§ 83. **Valve Diagrams.**—To determine the dimensions of the gearing driving the valves by means of eccentrics, the Müller-Reuleaux or Zeuner valve diagrams are generally used. A description of these diagrams will now be given. It should be noted that it is not necessary to consider the length of the eccentric rod, as this is generally made so long in proportion to the stroke of the eccentric, that the effect of the length of the rod may be neglected.

§ 84. **Müller-Reuleaux Diagram for ordinary D Slide Valves.**—Here—

a denotes breadth of port in valve face.
e_o and e_u denote outside lap at top and bottom respectively.
i_o „ i_u „ inside „ „ „ „
v_o „ v_u „ linear lead „ „ „
ϵ_o „ ϵ_u „ cut-off in the cylinder „ „

As a rule, when designing the valve gear, the breadth of port a, the linear lead of the valve top and bottom, and the cut-off are given. The amount of eccentricity is then determined, and taking it as the radius, the valve circle is described from the centre o. (See Fig. 160). Let AOB be the travel of the valve. Draw circles round the two dead points A and B with radii equal to the lead of the valve at the top and bottom of the cylinder respectively, and plot the cut-off AC as a percentage of the stroke of the valve for one side of the piston, say the top side. Then if an arc be drawn through C with radius equal to the length of the connecting rod, bearing the same ratio to the amount of eccentricity as the length of the

main connecting rod does to the crank, it will cut the circle of the valve at point 4. o4 will then represent the position of the crank corresponding to the cut-off on the top side of the cylinder. Through point 4 draw a tangent to the lead circle at the top of the stroke, and draw a parallel line touching the lead circle at the bottom of the stroke. If a third parallel line XY be then drawn through the centre o, and o

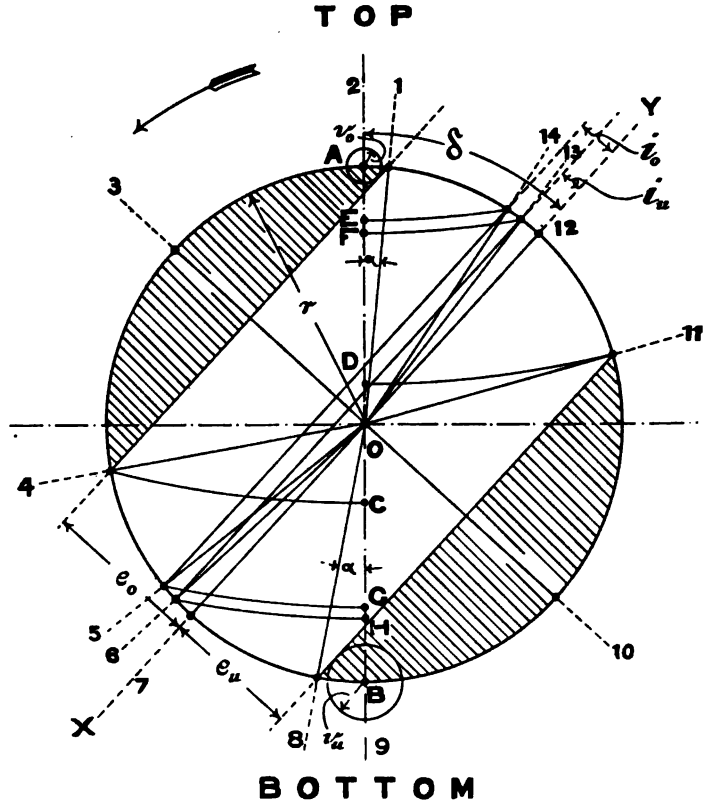


Fig. 160.

joined to the points where the first two lines cut the valve circle, o11 will represent the position of the crank corresponding to the cut-off at the bottom end, and the angular lead δ = the angle AOY. The outside lap of the valve is represented by e_o and e_u. The inside lap can be measured in the same way from the central line XY. If the inside lap is positive, it will be on the opposite side to the outside lap for the same end of the piston; if it is negative, it will be on the same side.

Principal Positions of the Crank. (See Diagram, Fig. 160.)

Position 1. Admission, top end.
„ 2. Crank at top of stroke, port open by amount equal to the lead $= v_o$.
„ 3. Maximum port opening, top end.
„ 4. Cut-off, top end.
„ 5. Beginning of exhaust, top end.
„ 7. Valve in middle position.
„ 14. Beginning of compression, top end.
„ 8. Admission, bottom end.
„ 9. Crank at bottom of stroke, port open by amount equal to the lead $= v_u$.
„ 10. Maximum port opening, bottom end.
„ 11. Cut-off, bottom end.
„ 12. Valve in middle position.
„ 13. Beginning of exhaust, bottom end.
„ 6. Beginning of compression, bottom end.

The diagram also shows—

AC : $2r = \epsilon_o$ = cut-off, top end.
BD : $2r = \epsilon_u =$ „ bottom end.
BG : $2r =$ exhaust lead, top end.
AE : $2r =$ compression, „
AF : $2r =$ exhaust lead, bottom end.
BH : $2r =$ compression, „

§ 85. **Zeuner's Valve Diagram** (Fig. 161).—This diagram is constructed in the following way:—Let AB be the direction of the travel of the valve, and O the centre of the diagram. Describe a circle with any given radius about O. Plot off from A the cut-off AC $= \epsilon_o \times$ AB, and draw the crank position O3 corresponding to this cut-off for the top end, taking the length of the connecting rod into account. Crank position O1 is then drawn, representing the point where the port opens on the top side. The angle 1O3 is bisected. The line bisecting it forms with the line MN (diameter of the circle AB) the angle of advance δ = angle MO2. With half the stroke of the eccentric as radius, the eccentric circles are then drawn through O, having their centres on the radii O2 and O6 respectively. For any given position of the crank OX, the line OY cut by the valve circle will be the distance of the valve from its middle position. The portions OP = OQ of the lines representing the crank positions O3 and O1 are respectively the outside lap, and the distance RS represents the lead of the valve at the top end. Let BG be the exhaust lead at this

end, then the crank position obtained from it (taking the length of the connecting rod into account) will be o4, and the corresponding inside lap $i_o = OT = OU$.

Crank position o5 is that in which compression begins at the top end. If the inside lap $OT = OU$ of the valve (top end) is positive, the circle drawn with centre o, taking the inside lap as the radius, must lie in the lower valve circle (as shown); if negative, it will lie in the upper valve circle, as indicated by the dotted line. In the latter case,

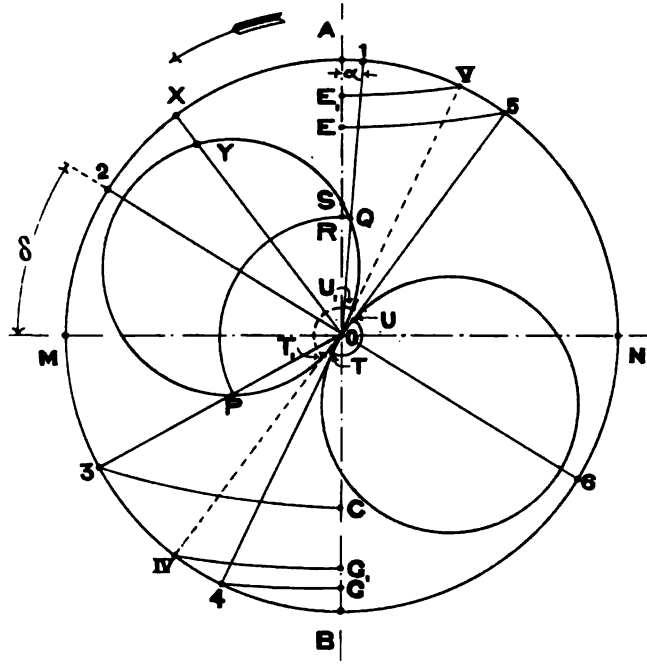


Fig. 161.

crank position o IV will correspond to the exhaust lead, and crank position o V to compression on the same side of the piston. The diagram for the bottom end is drawn in the same way.

§ 86. **Variations in the Cut-off.**—To vary the cut-off in a cylinder with link motion, the link itself is generally used when the cut-off is not less than about 45 %, and the variation required is only from 15 to 20 %. If a smaller cut-off is required, a separate expansion valve is generally provided (Meyer's or Rider's), but these valves are

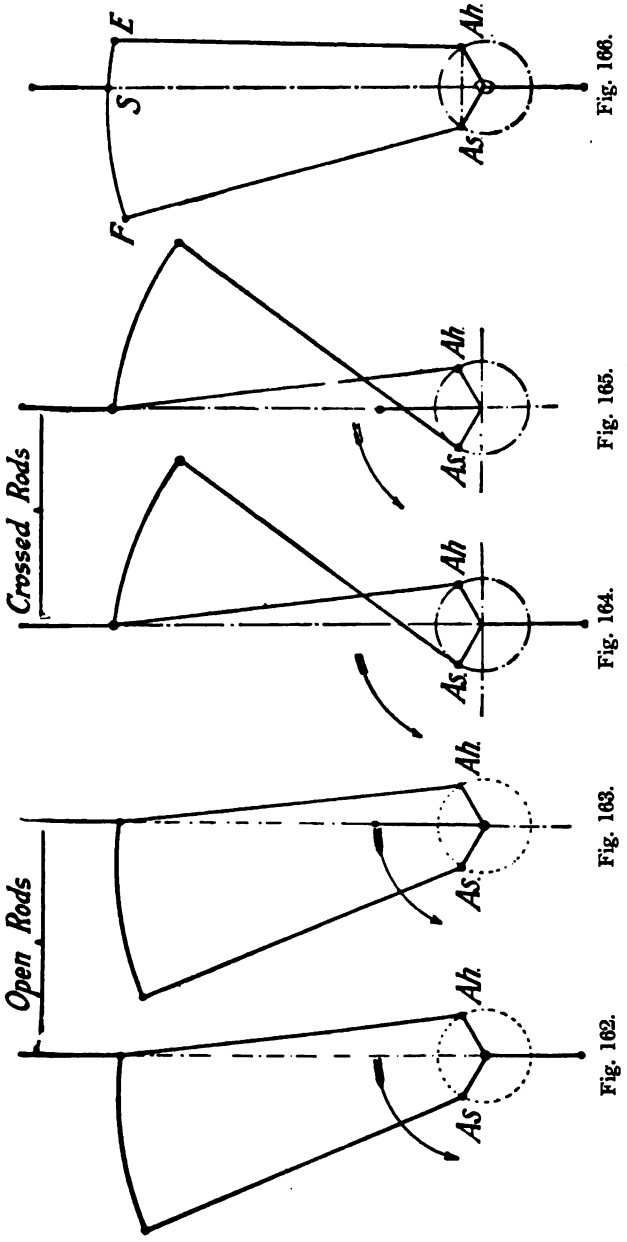


Fig. 162. Fig. 163. Fig. 164. Fig. 165. Fig. 166.

seldom used in marine engines, and need not therefore be further considered here.

§ 87. **Stephenson's Link Motion.**—In this form of motion there are two eccentrics, one for the ahead and the other for the astern gear. The amount of eccentricity and the angle of lead are

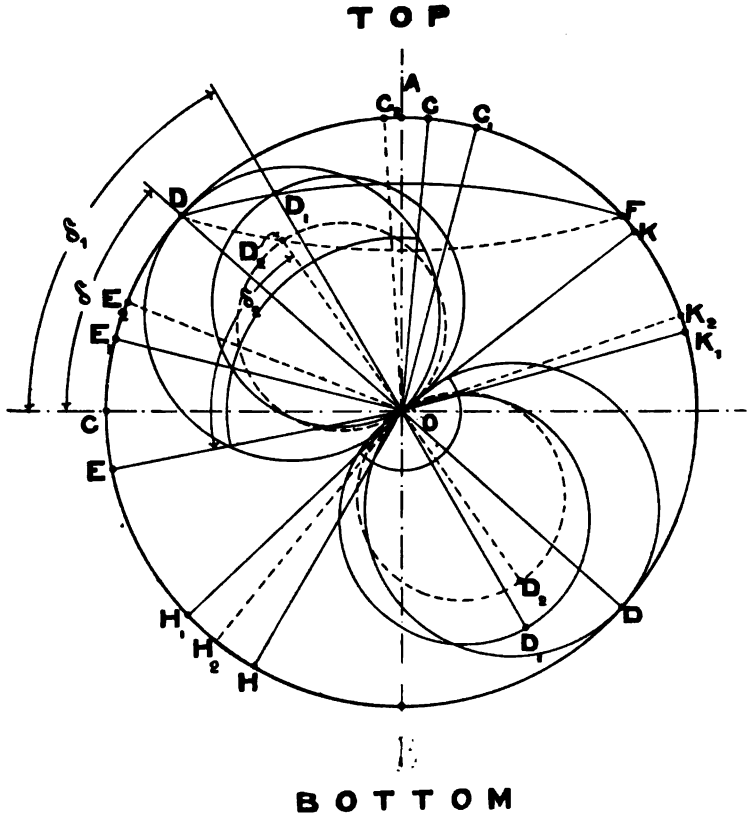


Fig. 167.

determined from the valve diagram for the maximum cut-off, and are as a rule the same both for the ahead and the astern gear. As regards the method of connecting the eccentric rods to the link, a distinction is made between *open rods* (Figs. 162 and 163) and *crossed rods* (Figs. 164 and 165). When both eccentrics are so placed that their eccentricity is towards the link, the difference between the two methods is clearly seen.

If the gearing lies fully over to one side, only that eccentric which has the upper end of its rod in line with the head of the valve rod need be considered. If the link is only partly over to one side, both eccentrics act, their effect being to reduce the cut-off, and vary the lead, compression, and exhaust lead. With open eccentric rods these will be increased, and with crossed eccentric rods they will be diminished. In both cases, if the link motion is only partly over, the maximum port openings will be reduced, but, for the same positions of the link, the port openings will be larger with open than with crossed rods.

The various functions of the link motion, between the two full-over positions, may be approximately determined by graphical methods as follows :—

In Fig. 167 let AB be the path of the valve rod, OD the amount of eccentricity, $COD = \delta$, the angle of advance. Through D describe the arc of a circle DF having its centre on the line AB, and a radius x which can be calculated from Fig. 166 by means of the equation $x = \frac{Ah.E \times Ah.AS.}{2EF}$. If the eccentric rods are open (not crossed), the concave side of this arc is turned towards the central point O as shown ; if they are crossed, the convex side will be turned towards O, as shown by the dotted lines. Upon the arc DF the lengths DD_1 and DD_2 calculated from $DD_1 = DD_2 = DF \times \frac{ES}{EF}$ are then marked off. OD_1 will then be the amount of eccentricity, and δ_1 the angle of advance of an eccentric under these conditions, which corresponds to the farthest point S reached by the motion of the valve rods with open eccentric rods. OE_1 will be the position of the crank when expansion begins. The lead, exhaust lead, and compression are determined by the method already described. If the rods are crossed, OD_2 will be the resultant amount of eccentricity, and δ_2 the corresponding angle of advance.

The corresponding positions of the crank are :—

	Full Gear.	Link Motion. Open Rods.	Link Motion. Crossed Rods.
Admission - - -	OG	OG_1	OG_2
Beginning of expansion -	OE	OE_1	OE_2
,, exhaust lead -	OH	OH_1	OH_2
,, compression -	OK	OK_1	OK_2

Various Types of Valve Gear.

The following systems of valve gear, as well as Stephenson's link motion, are frequently used in marine engines:—

§ 88. **Klug Valve Gear.**—The motion of the valve is obtained by an eccentric lying in the same direction as the main crank (see Fig. 168). The centre line of the eccentric rod motion OU is as a rule laid down at right angles, or nearly so, to the piston and valve rods. If it is necessary, from want of space, to arrange it otherwise, the eccentric must be at the same angle as the crank, and the motion must be transmitted from the eccentric rod to the valve rod by a suitable arrangement of levers.

In Fig. 168, O is the crank shaft, U the reversing shaft, OK the crank, $OE_1 = r$ amount of eccentricity. The usual proportions of the parts of the valve gear are—

$$\begin{aligned} OU &= \text{from } 5 \text{ to } 7r. \\ \text{Eccentric rod } E_1U &= \quad,, \quad \sqrt{r^2 + (OU)^2} \text{ to } OU. \\ A_1U &= \text{about } 0{\cdot}5\ E_1U. \\ CU &= \quad,, \quad 6 \text{ to } 8r. \end{aligned}$$

The angle of throw α of the reversing shaft from its centre is generally about 15° to 20°. By making this angle smaller the cut-off is diminished, and compression and exhaust lead increased; by reversing the process, opposite conditions will be obtained.

The lead of the valve will be constant if $E_1U = \sqrt{r^2 + OU^2}$.

The distribution of steam can best be determined by the use of a small model; but, failing this, it can be worked out graphically.

The curve described by the end point A of the eccentric rod may be drawn for different positions of the reversing lever UC. The circle described by the eccentric is then divided into eight to twelve equal parts, and for each of these positions the relative position of point A is determined. By joining these various points, the desired curve for any given position of the reversing lever UC is found. If from positions A_1 and A_5 of the end of the eccentric rod, corresponding to the dead points of the crank, the leads v_o and v_u respectively be plotted parallel to the path of the valve rod, and an arc drawn through these points, with AD

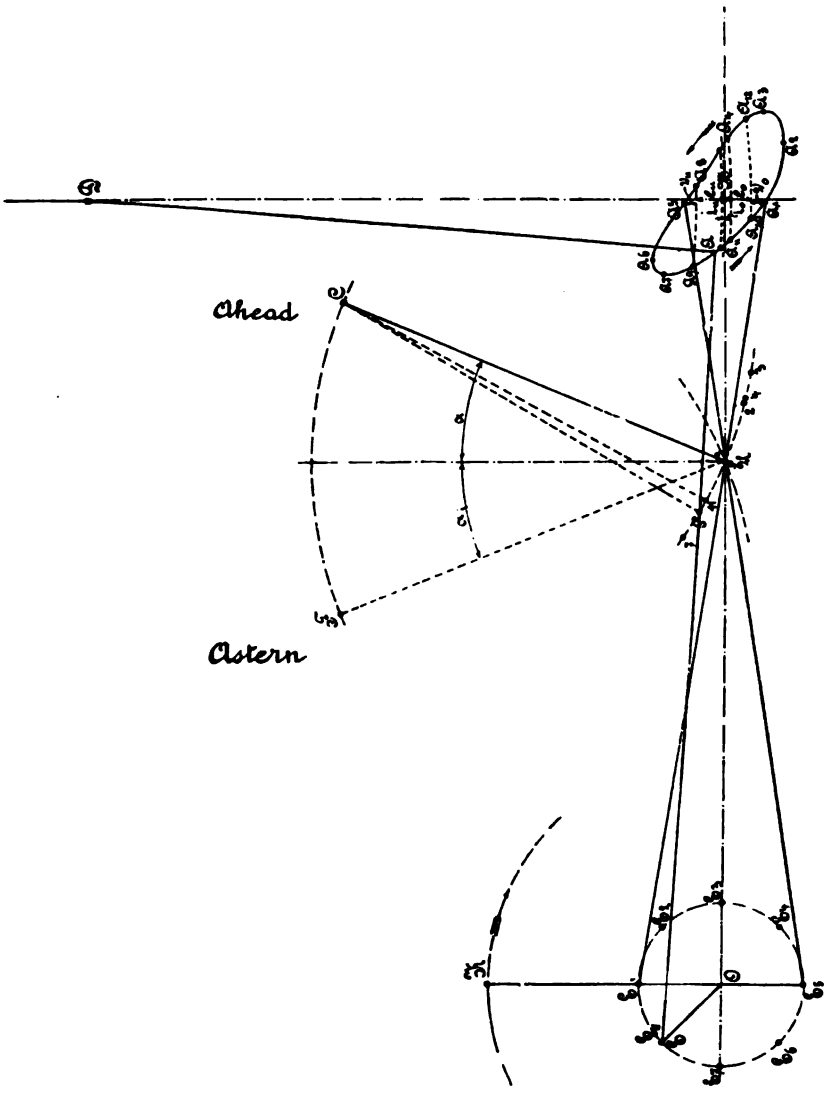


Fig. 168.

as the radius, then, as M corresponds to the middle position of the valve—

e_o will be = outside lap at the top end.

e_u ,, = ,, ,, bottom end.

M

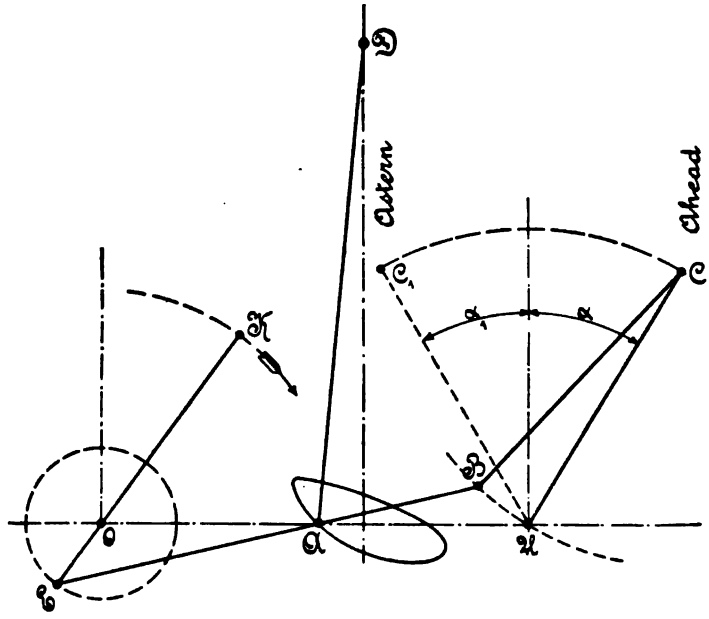


Fig. 169.

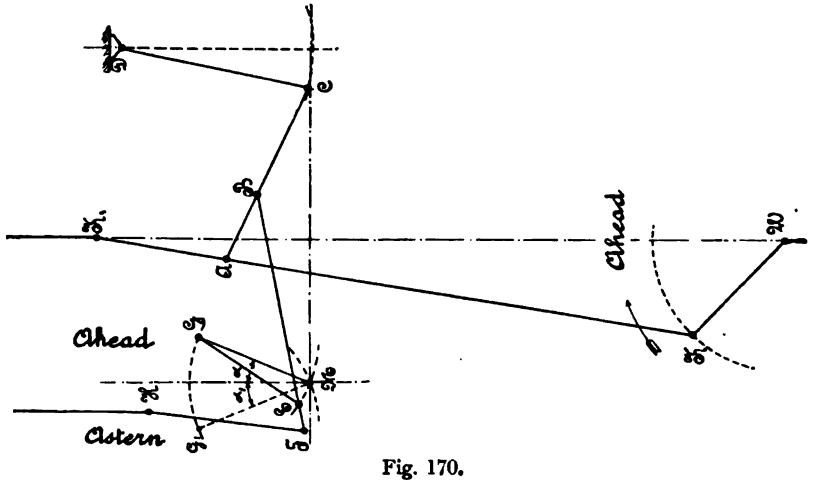


Fig. 170.

The inside lap of the valve is determined from the positions assigned to the crank or piston at the beginning of the exhaust lead, at the top and bottom ends respectively. Further, the points on the eccentric circle

corresponding to the positions at the end of the eccentric rod are as follows :—

A_8	beginning of	admission	at the	top end.
A_9	,,	expansion	,,	,,
A_{10}	,,	admission	,,	bottom end.
A_{12}	,,	expansion	,,	,,

From these points, by reversing the construction of the diagram, the corresponding positions of the crank and of the piston respectively can be found. For instance, position of the eccentric E_4, at which exhaust begins at the upper end of the cylinder, corresponds to position A_4 of the end of the eccentric rod. If an arc be drawn through A_4 with AD as radius, it will cut the curve on the other side at the point A_{11}, corresponding to position E_{11} of the eccentric, at which compression begins at the top end. The distance between the two arcs drawn through M and A_4 gives the inside lap i_o. The inside lap i_u for the bottom end is found in the same way. It should also be noted that the inside lap is negative, if it is on the same side as the outside lap for the same end of the cylinder. The valve motion is reversed by moving the shaft, and the lever connected to it, from position UC to position UF.

§ 89. **Marshall's Valve Gear.**—This is very similar to the Klug valve gear, except that the valve connecting rod is not at the end of the eccentric rod, but comes in between points B and E. (See Fig. 169.) The eccentric forms an angle of 0° or of 180° with the crank. The steam distribution is determined in the same way as has already been fully described for the Klug valve gear.

§ 90. **Joy's Valve Gear.**—Motion is here communicated to the valve from a point on the main connecting rod. (See Fig. 170.) The movement of the valve is regulated in the same way as in the Klug or Marshall gear. It is best determined from a small model, or by drawing the curve described by the point F. The usual proportions of the parts of this valve gear, as shown in Fig. 170, are—

$\frac{K_1A}{K_1K}$	= 0·20 to 0·23	$\frac{BE}{KW}$	= 1·5 to 1·65
$\frac{AC}{KW}$	= 1·4 ,, 1·5	$\frac{EF}{KW}$	= 0·2 ,, 0·3
$\frac{BC}{AC}$	= 0·5 ,, 0·6	$\frac{EG}{KW}$	= 0·8 ,, 1·0
$\frac{CD}{KW}$	= 1·4 ,, 1·5	$\frac{FH}{KW}$	= 1·0 ,, 2·5

At the top and bottom of the stroke, points E and M coincide, from which it follows that *the lead of the valve is constant for each position*

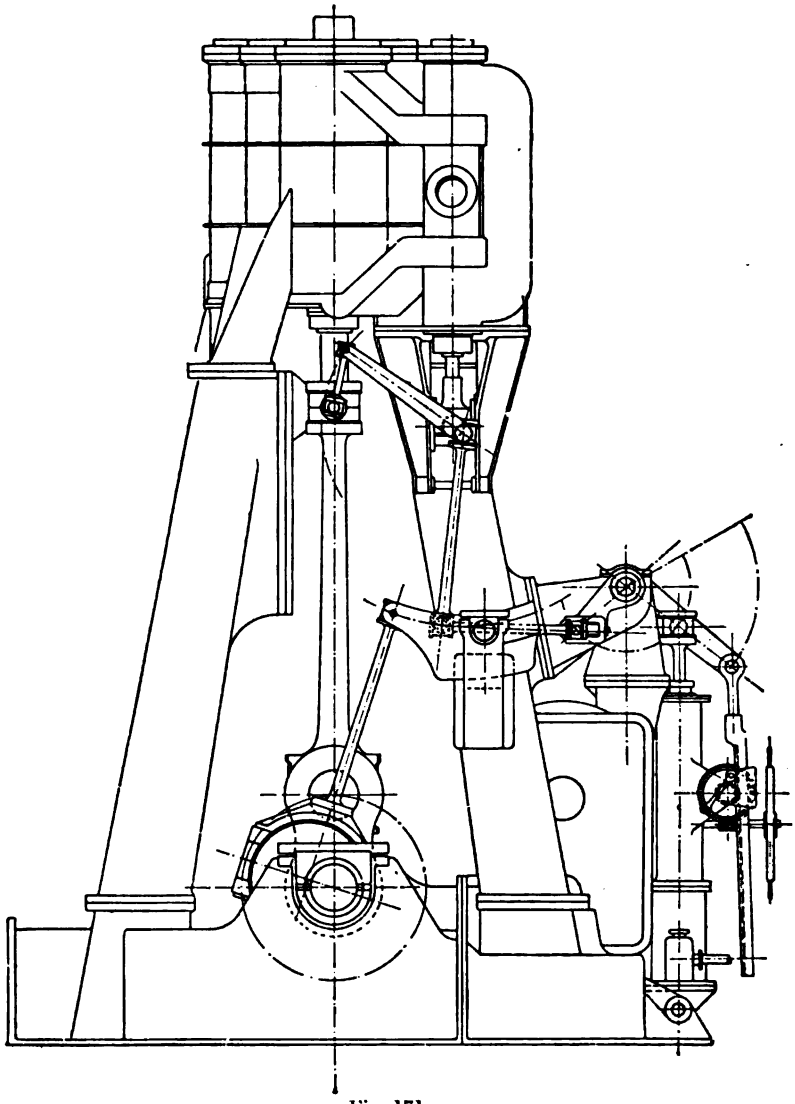


Fig. 171.

of the reversing lever MG. By increasing the angle of throw of this lever the cut-off is increased, and the exhaust lead and compression

diminished, and *vice versâ*. The engine is reversed by shifting the lever MG into the position MG_1. The angles α and α_1 may be about 15° to 20°.

§ 91. **Heusinger Valve Gear.**—In this valve motion the valve is also driven by an eccentric on the crank shaft, and by a lever connected to the crosshead of the engine. (See Fig. 171.) The eccentric works on one end of a slot link which can rotate or swing about its centre. The radius of the link being equal to the length of the valve connecting rod, the link is in its central position when the piston is at the top and bottom of the stroke. The upper end of the valve connecting rod is fixed to a lever, connected to the valve rod, and also at the other end, through a short link, to the main crosshead. The motion of the engine is reversed by moving the end of the valve connecting rod from one end of the slide link to the other. This is done by a reversing shaft common to all the cylinders, which shifts the link block by means of a reversing lever and link, in the same way as in the ordinary link motion. The cut-off may be varied by moving the link block nearer to the centre of the reversing link. The lead is constant for all positions of the slide block.

Piston Rods.

§ 92. **Maximum Load.**—The so-called *maximum load* is generally taken as a basis for calculating the strength of the following parts, viz., the piston, piston rod, crosshead, connecting rod, and crank shaft. This load P = area of the H.P. piston × boiler pressure. The rods of all the cylinders in compound engines are calculated on this basis. They are all made alike, to avoid the multiplication of spare parts, and to promote interchangeability.* If the rod is driven by an L.P. cylinder with an H.P. cylinder above it, the maximum load is taken at P = area of H.P. cylinder × pressure in boiler (above atmosphere), + area of L.P. cylinder × maximum pressure of steam entering L.P. cylinder in pounds per square inch (absolute). The highest admission pressure in the L.P. cylinder in the triple-expansion engines here considered may be taken at 50 lb. per square inch (absolute), and in quadruple-expansion engines at 30 lb. per square inch (absolute).

Example.—In a quadruple-expansion engine let each of the two middle cranks be driven by an L.P. and an H.P. cylinder placed "tandem,"

* The same interchangeability is provided for when the H.P. cylinder rests upon one of the other cylinders.

while the M.P. cylinders are arranged over the two outside cranks. Diameter of the H.P. cylinder (800 mm.) = 30 inches; of the L.P. cylinder (2400 mm.) = 90 inches. Pressure in boiler (15 atmospheres), 200 lb. per square inch. The maximum load on the rods in pounds will then be—

$$P = (\cdot 78 \times 30^2 \times 200) + (\cdot 78 \times 90^2 \times 30) = 337{,}940 \text{ lb.} = 150{\cdot}8 \text{ tons.}$$

This load P forms the basis for calculating all the four sets of rods, &c., including those of the M.P. cylinders.

Pistons.

§ 93. **General Remarks.**—In most large engines, and in the main engines of all large warships, the pistons are made of *cast steel.* In

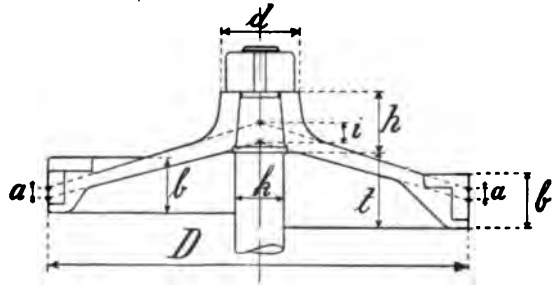


Fig. 172.

vessels where weight is no object (cargo steamers, &c.) they are made of *cast iron*, and in very light ships (torpedo-boats) of *forged steel.*

§ 94. **Cast-steel Pistons.**—Typical shapes are shown in Figs. 172 to 174. Thickness of boss, $d = 1{\cdot}5$ to $1{\cdot}7k$. $1{\cdot}7k$ should be the value used for small pistons, and those of engines where weight is no object; $1{\cdot}5k$ for large pistons and light engines. Height of boss, $h = 1{\cdot}1k$.

The thickness of the steel piston in the middle i is obtained from the formula $i = \kappa \times c$. The value of κ is given in Table No. 19; the coefficient c is taken at—

$c = 1$	for flat pistons, or those in which the inclination measured on the inside is very slight (angle 0° to 6°).
$c = 0{\cdot}85$ to $0{\cdot}95$	,, slightly coned pistons (angle 6° to 18°).
$c = 0{\cdot}75$,, $0{\cdot}85$	,, medium coned pistons (angle 18° to 28°).
$c = 0{\cdot}65$,, $0{\cdot}75$	,, strongly coned pistons (angle 28° to 35°).

The thickness a near the outer circumference is obtained from the equation $a = 0{\cdot}45i$ to $0{\cdot}55i$. It is best not to make the piston quite flat, but slightly conical. In engines with several cylinders the dimensions t, h, and b (Fig. 172) are generally made the same for the H.P., M.P., and L.P. cylinders. If an inclination of 1 : 5·5 to 1 : 6·5 be taken for the internal slope of the largest piston, normal proportions for all the pistons will be obtained.

In Table No. 19 the steam pressures are to be taken as absolute admission pressures, and the values given cannot be used for pistons which have an internal slope of more than 35° to the horizontal.

Fig. 172A. *Safety Nut*

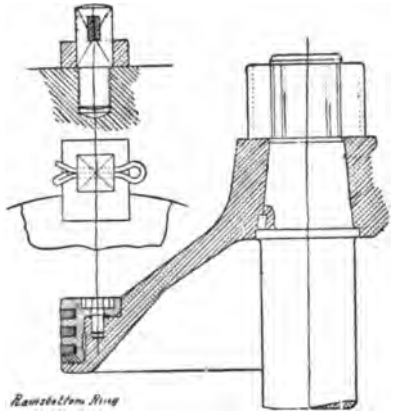


Fig. 173.

TABLE No. 19.

Thickness of Steel Pistons. Values of K.

(Pressure in pounds per square inch absolute.)

Diameter of Cylinder.	Pressures, lb. per sq. inch, 15 to 30.	Pressures, lb. per sq. inch, 30 to 55.	Pressures, lb. per sq. inch, 55 to 85.	Pressures, lb. per sq. inch, 85 to 114.	Pressures, lb. per sq. inch, 114 to 142.	Pressures, lb. per sq. inch, 142 to 170.	Pressures, lb. per sq. inch, 170 to 200.	Pressures, lb. per sq. inch, 200 to 227.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
15 to 23	1	1 1/4	1 3/8	1 1/2	1 3/4	2	2 1/8	2 1/2
23 ,, 31	1 3/8	1 1/2	1 3/4	2	2 1/2	2 3/4	3	3 3/8
31 ,, 39	1 1/2	1 3/4	2	2 1/2	3 1/4	3 1/2	3 3/4	4
39 ,, 47	1 3/4	2	2 3/8	3	3 1/2	4	4 1/8	...
47 ,, 55	2	2 1/8	2 3/4	3 1/8	3 3/4	4 1/8	...	...
55 ,, 63	2 3/8	2 1/2	3 1/4	3 1/2	4	4 3/8	...	...
63 ,, 71	2 1/2	3	3 3/8	4	4 3/8	...	...	...
71 ,, 79	2 3/4	3 1/4	3 1/2	4 1/8	...	...	...	...
79 ,, 86	3	3 3/8	3 3/4	4 3/8	...	...	...	...
86 ,, 94	3	3 1/2	4	...	...	...	...	...
94 ,, 102	3 1/8	3 1/2	...	...	...	...	...	...
102 ,, 110	3 3/8	3 3/4	...	...	...	...	...	...

§ 94A. **Cast-iron Pistons.**—For typical form, see Figs. 175, 176, 179, and 180.

Thickness of boss, $d = 1{\cdot}5$ to $1{\cdot}7k$.

Height of boss, $h = 1{\cdot}4k$.

The thickness of the boss is the same as for steel pistons, because it is strengthened by the upper and lower wall.

Depth of piston = about height of boss.

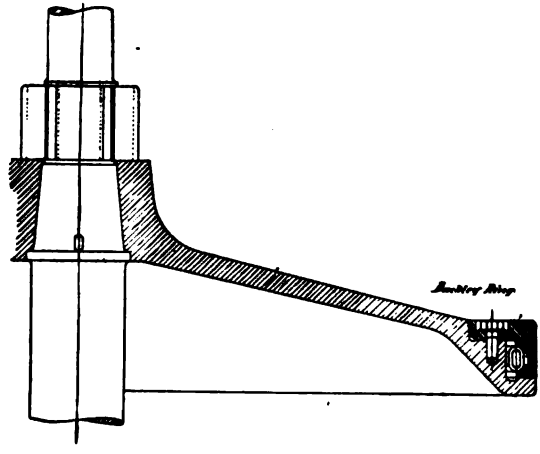


Fig. 174.

Thickness δ of the upper and lower walls and of the ribs should be from about $\frac{D}{60}+\cdot4$ inch to about $\frac{D}{40}+\cdot4$ inch. The higher values are for the H.P. and M.P., the lower for the L.P. piston. The number of ribs are as follows:—

$z=$ 4	for 12 to 24	inches	diameter	of	cylinder.
$z=$ 6	„ 24 „ 40	„	„	„	
$z=$ 8	„ 40 „ 60	„	„	„	
$z=$ 10-12	„ 60 „ 80	„	„	„	

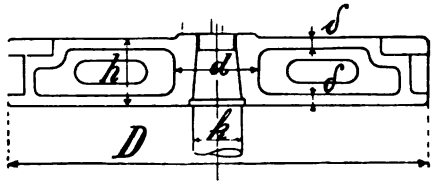


Fig. 175.

The walls or surfaces between the ribs or webs must be sufficiently strong to resist the maximum absolute pressure to which the piston is exposed. If the strain is too great, the above number of ribs should be increased. For different methods of fitting the rod into the piston, see " Piston Rods," page 191.

§ 95. **Piston Packing.**—1. *Ramsbottom Rings* may be used for all pressures. The rings are almost always of cast iron. The inner and

outer circumferences are eccentric in regard to one another. The rings are cut across diagonally at the thinnest part, so that no scoring of the cylinder can take place where the ends meet. The rings are turned in the first instance of somewhat larger diameter than the cylinder; in most cases a cylindrical casting is first made, from which the rings

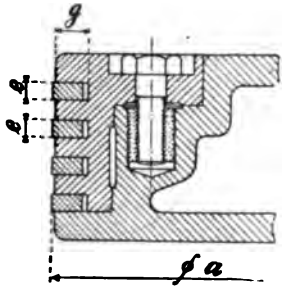


Fig. 176.

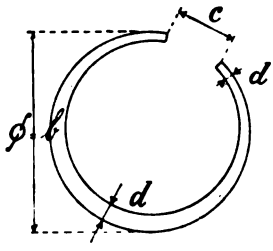


Fig. 177.

TABLE No. 20.

Ramsbottom Piston Rings. (See Figs. 176 and 177.)

a Bore of Cylinder.	*b* Diameter of Ring before Stretching.	*c* Length of Piece Cut Out.	*d* Thickness of Ring.	*e* Depth of Ring.	*f* Number of Rings.	*g* Depth of Groove in Piston.
Inches.	Inches.	Inches.	Inches.	Inches.		Inches.
12	$12\frac{3}{8}$	$1\frac{9}{16}$	$\frac{1}{2}$ to $\frac{9}{16}$	$\frac{11}{16}$ to $\frac{13}{16}$	2 to 3	$\frac{11}{16}$
15	$15\frac{9}{16}$	$2\frac{3}{16}$	$\frac{9}{16}$ „ $\frac{5}{8}$	$\frac{11}{16}$ „ $\frac{13}{16}$	2 „ 3	$\frac{3}{4}$
20	$20\frac{13}{16}$	$2\frac{3}{4}$	$\frac{5}{8}$ „ $\frac{11}{16}$	$\frac{13}{16}$ „ 1	2 „ 3	$\frac{13}{16}$
24	25	$3\frac{1}{8}$	$\frac{11}{16}$ „ $\frac{3}{4}$	$\frac{13}{16}$ „ 1	3	$\frac{15}{16}$
27	$28\frac{3}{16}$	$3\frac{9}{16}$	$\frac{3}{4}$ „ $\frac{7}{8}$	$\frac{13}{16}$ „ 1	3	1
30	$31\frac{3}{16}$	$4\frac{1}{8}$	$\frac{13}{16}$ „ 1	$\frac{13}{16}$ „ 1	3 to 4	$1\frac{1}{8}$
36	$37\frac{3}{8}$	$4\frac{3}{4}$	$\frac{7}{8}$ „ $1\frac{1}{8}$	$\frac{13}{16}$ „ 1	3 „ 4	$1\frac{1}{4}$
39	$40\frac{9}{16}$	$5\frac{1}{8}$	$\frac{15}{16}$ „ $1\frac{3}{16}$	$\frac{13}{16}$ „ 1	3 „ 4	$1\frac{5}{16}$
42	$43\frac{13}{16}$	$5\frac{1}{2}$	$\frac{15}{16}$ „ $1\frac{1}{4}$	1	3 „ 4	$1\frac{3}{8}$
48	50	$6\frac{1}{8}$	$\frac{15}{16}$ „ $1\frac{5}{16}$	1	3 „ 4	$1\frac{7}{16}$
51	53	$6\frac{1}{2}$	$\frac{15}{16}$ „ $1\frac{7}{16}$	1	3 „ 4	$1\frac{9}{16}$
54	$56\frac{3}{16}$	$7\frac{1}{16}$	$\frac{15}{16}$ „ $1\frac{7}{16}$	1	3 „ 4	$1\frac{9}{16}$
60	$62\frac{3}{8}$	$7\frac{1}{2}$	$\frac{15}{16}$ „ $1\frac{1}{2}$	1	3 „ 4	$1\frac{5}{8}$
63	$65\frac{9}{16}$	$8\frac{1}{16}$	1 „ $1\frac{9}{16}$	1	3 „ 4	$1\frac{11}{16}$
66	$68\frac{3}{4}$	$8\frac{7}{16}$	1 „ $1\frac{9}{16}$	1	3 „ 4	$1\frac{11}{16}$
72	$74\frac{15}{16}$	$9\frac{1}{16}$	1 „ $1\frac{5}{8}$	$1\frac{1}{16}$	3 „ 4	$1\frac{3}{4}$
75	$78\frac{1}{8}$	$9\frac{13}{16}$	1 „ $1\frac{5}{8}$	$1\frac{1}{16}$	3 „ 4	$1\frac{3}{4}$
78	$81\frac{1}{8}$	$10\frac{7}{16}$	1 „ $1\frac{5}{8}$	$1\frac{1}{16}$	3 „ 4	$1\frac{3}{4}$

are parted off. A piece is then cut out of the ring, and the ring is pulled in with a band or soldered together, and turned to fit the bore of the cylinder. Larger rings are turned in the first instance to the correct size of the cylinder, and after cutting out the piece, are fitted into the cylinder, and carefully hammered to a true circle. For dimensions of the rings, &c., see Table No. 20.

In H.P. cylinders of smaller diameter four rings are used, which need not be made as deep as the dimensions given in the table. With small pistons a number of small steel Ramsbottom rings are often used; say, about three rings from $\frac{1}{4}$ to $\frac{5}{16}$ inch deep, and about $\frac{5}{16}$ to $\frac{1}{4}$ inch thick for pistons of 7 to 10 inches diameter.

In pistons with Ramsbottom rings care must be taken that the piston does not run too far into the coned portion at either end of the cylinder, so as to allow the outer ring to become jammed.

2. A ring very generally used for L.P. cylinder pistons is the *Buckley Ring* (Fig. 178). A flattened helical steel spring is so placed, behind the inner slanting surfaces of two packing rings, that it presses them out

TABLE No. 21.

Buckley Piston Packing. (See Fig. 178.)

a Diameter of Cylinder.	*b* Depth of Ring.	*c* Thickness of Ring.	*d* Length of Piece Cut Out.	*e* Thickness of Ring at Bottom.
Inches.	Inches.	Inches.	Inches.	Inches.
$20\frac{1}{2}$ to $22\frac{1}{2}$	$1\frac{7}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{15}{16}$
$22\frac{1}{2}$,, $24\frac{1}{2}$	2	$\frac{5}{8}$	$1\frac{9}{16}$	1
$24\frac{1}{2}$,, $27\frac{1}{2}$	$2\frac{1}{8}$	$\frac{11}{16}$	$1\frac{11}{16}$	$1\frac{1}{16}$
$27\frac{1}{2}$,, $30\frac{1}{2}$	$2\frac{1}{4}$	$\frac{11}{16}$	$1\frac{13}{16}$	$1\frac{1}{8}$
$30\frac{1}{2}$,, $33\frac{1}{2}$	$2\frac{3}{8}$	$\frac{11}{16}$	$1\frac{7}{8}$	$1\frac{3}{16}$
$33\frac{1}{2}$,, $36\frac{1}{2}$	$2\frac{1}{2}$	$\frac{11}{16}$	$1\frac{15}{16}$	$1\frac{1}{4}$
$36\frac{1}{2}$,, $39\frac{1}{2}$	$2\frac{5}{8}$	$\frac{11}{16}$	$2\frac{1}{16}$	$1\frac{5}{16}$
$39\frac{1}{2}$,, $43\frac{1}{2}$	$2\frac{3}{4}$	$\frac{3}{4}$	$2\frac{1}{8}$	$1\frac{3}{8}$
$43\frac{1}{2}$,, $47\frac{1}{2}$	$2\frac{7}{8}$	$\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{7}{16}$
$47\frac{1}{2}$,, 51	3	$\frac{3}{4}$	$2\frac{3}{8}$	$1\frac{1}{2}$
51 ,, 55	$3\frac{1}{8}$	$\frac{3}{4}$	$2\frac{7}{16}$	$1\frac{9}{16}$
55 ,, 59	$3\frac{1}{4}$	$\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{5}{8}$
59 ,, 67	$3\frac{3}{8}$	$\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{11}{16}$
67 ,, 75	$3\frac{1}{2}$	$\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{3}{4}$
75 ,, $82\frac{1}{2}$	$3\frac{5}{8}$	$\frac{7}{8}$	$2\frac{7}{8}$	$1\frac{13}{16}$
$82\frac{1}{2}$,, $90\frac{1}{2}$	$3\frac{3}{4}$	$\frac{7}{8}$	3	$1\frac{7}{8}$
$90\frac{1}{2}$,, $98\frac{1}{2}$	$3\frac{7}{8}$	$\frac{7}{8}$	$3\frac{1}{8}$	$1\frac{15}{16}$
$98\frac{1}{2}$ and upwards	4	$\frac{7}{8}$	$3\frac{1}{4}$	2

against the walls of the cylinder, and also up and down against the junk ring and the body of the piston. The thickness of the rings c is the same throughout. For dimensions of the Buckley ring see Table No. 21.

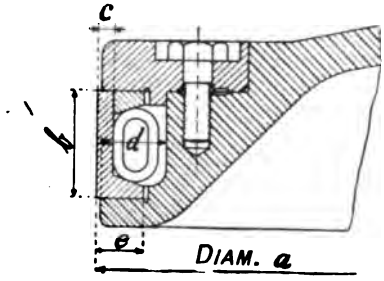


Fig. 178.

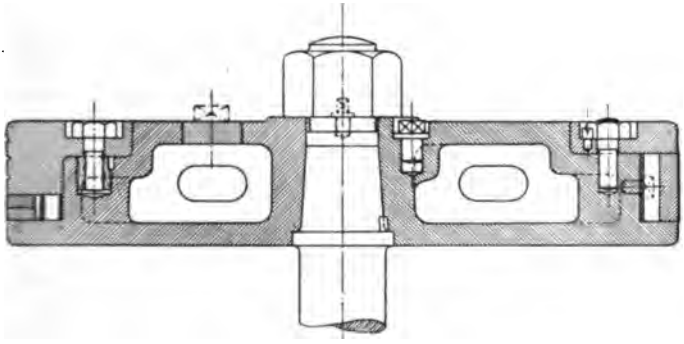


Fig. 179.

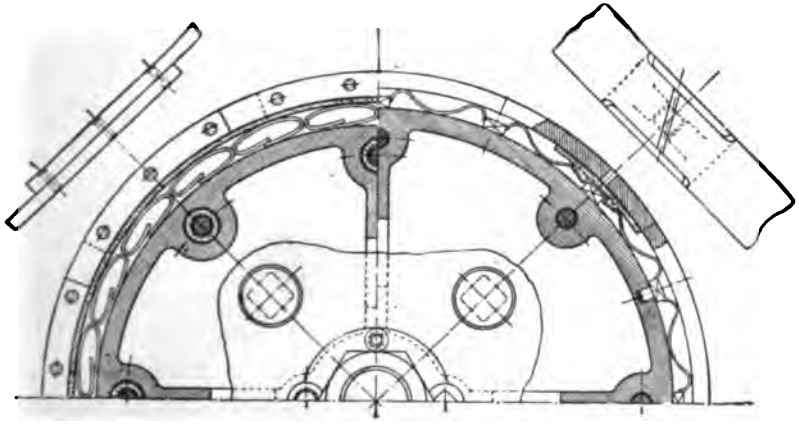


Fig. 180.

3. The *Peck* piston packing is often used for the L.P. and M.P. cylinder pistons. It consists chiefly of two cast-iron rings, in several sections, lying one above the other, which are pressed outwards against the sides of the cylinder by springs. One section in the upper and one section in the lower ring are riveted together, so that the steam cannot blow directly through the joint. Behind these sections lie bronze rings,

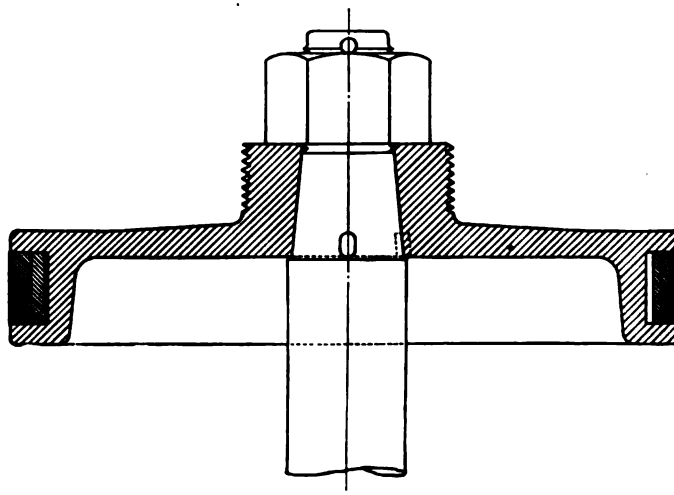


Fig. 181.

against which the springs bear. (Compare Figs. 179, 180, left half.) Above the cast-iron rings is a thick covering ring with turned grooves. Thickness of cast-iron segments, $\frac{9}{16}$ to $\frac{3}{4}$ inch. This packing is used chiefly for pistons of more than 4 feet diameter. In the latest types of the Peck piston, the springs are replaced by a self-adjustiug cast-iron ring.

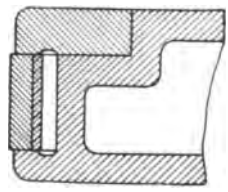


Fig. 182.

For both high and low pressures the following packings, which are not self-adjusting, are used :—

1. *Packing Ring, with Single Flat Springs.* —Behind the flat, cast-iron piston ring (of which there are sometimes two) lies a flat steel spring, which is made thinner towards either end. This type of packing is much used for donkey engines. (See Figs. 181 and 182.)

2. *Packing Ring, with Multiple Flat Carriage Springs.*—This old method of packing the piston, which is very convenient for adjusting and for repairs, has the disadvantage, that the springs are in constant

motion while the engine is running, and therefore apt to wear themselves, as well as the body of the piston, away. The height of the cast-iron rings is about the same as that of the Buckley ring; for high pressures the thickness of the rings should be somewhat more. (See Figs. 179, 180, right half.)

General Remarks on Piston Packings.—To prevent steam leaking past them, the smaller and thinner rings are often cut, as shown in Fig. 183. Larger rings are fitted with small plates overlapping the joint (Fig. 180, right hand). These are generally riveted to one end of the ring, and then fitted on to the other end. To reduce the wear and risk of fracture to a minimum, the piston ring should project as little as possible beyond the body of the piston. With small pistons of about 3 feet diameter, it is sufficient to turn the body of the piston about $\frac{1}{64}$ to $\frac{1}{32}$ inch smaller than the bore of the cylinder; but with pistons of larger diameter, the allowance should be from $\frac{1}{16}$ to $\frac{1}{8}$ inch.

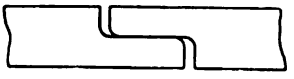


Fig. 183.

§ 96. **Clearance between the Piston and the Top and Bottom Covers of the Cylinder.**—As a rule this space is made larger between the piston and the bottom cover, than between the piston and top cover, because the wear of the bearings causes the piston, in course of time, to drop slightly. A larger clearance is allowed for unmachined cast-iron pistons than for turned cast-steel pistons. Sometimes the covers (especially if of cast steel) are turned on the inside; in any case it is advisable, if the cover and body of the piston are not turned, to check or test them in the lathe, in case there should be any undue thickening on one side, which would necessitate their being machined during erection, in order to give the necessary clearance.

TABLE No. 22.

Piston Clearances.

Diameter of Cylinder.	Clearance at Bottom.	Clearance at Top. (Thickness of Packing for Jointing Cylinder Cover included.)
Inches.	Inch.	Inch.
15 to 24	$\frac{1}{4}$ to $\frac{3}{8}$	$\frac{3}{16}$ to $\frac{1}{4}$
24 „ 39	$\frac{5}{16}$ „ $\frac{1}{2}$	$\frac{1}{4}$ „ $\frac{3}{8}$
39 „ 60	$\frac{1}{2}$ „ $\frac{9}{16}$	$\frac{5}{16}$ „ $\frac{1}{2}$
60 „ 78	$\frac{9}{16}$ „ $\frac{11}{16}$	$\frac{1}{2}$ „ $\frac{9}{16}$
78 „ 98	$\frac{11}{16}$ „ $\frac{13}{16}$	$\frac{9}{16}$ „ $\frac{5}{8}$
Above 98 inches	$\frac{13}{16}$ „ $\frac{7}{8}$	$\frac{5}{8}$ „ $\frac{11}{16}$

§ 97. **Thickness of Junk Rings.**—This should be such that the junk ring, when removed (Fig. 182), is strong enough not to bend or break. If the junk ring is made of cast iron, a thickness of $1\frac{1}{8}$ to $1\frac{1}{2}$ inch is sufficient in pistons of less than 3 feet in diameter; in larger pistons the thickness should be $1\frac{1}{2}$ to $2\frac{3}{8}$ inches. If the ring is of steel, the thickness may be less; but as the bolts are generally made with countersunk heads, there is not much saving in weight by using steel. Holes should be tapped in these rings for starting pins and lifting bolts.

Junk Ring Pins.—These are generally made in the form of tap bolts. The thread of the screw is as strong as possible, and the heads are either square or hexagonal. Cheese-headed bolts must not be used, as the frequent screwing up and unscrewing spoils the slits.

Size of Junk Ring Pins.

For cylinder diameters of from 15 to 24 inches, about $\frac{7}{8}$ to 1 inch.
" " " 24 " 39 " " 1 " $1\frac{1}{8}$ "
" " " 39 and over, " $1\frac{1}{8}$ " $1\frac{1}{4}$ "

Their pitch should be—

For the H.P. cylinder = 5 to 7 times their diameter.
" M.P. " = 6 " 8 " "
" L.P. " = 7 " 10 " "

To prevent the set screws of the junk ring rusting into the body of the piston, they are often screwed into gunmetal plugs, which are themselves screwed into the piston with a fine thread. Some simple means must be used to lock the bolts, such as a screw let in close to the head, to prevent the latter slacking back. Sometimes a brass washer is placed below the head of the set screw, which, as it expands with the heat, keeps it tight.

§ 98. **Remarks.**—All set screws and lock nuts on the piston must be very strong, and of such dimensions that they do not wear with the constant screwing and unscrewing. The piston rings must be a perfect fit between the junk ring and the body, especially if the surface they present to both is small. The rings must also be a steam-tight fit against the walls of the cylinder. To draw the piston off the rod, two large tapped holes are made in the body. Studs are fitted into these, by means of which a bar or clamp can be screwed down on to the top of the rod, and the piston slacked off and lifted from it. Pistons which fit very tightly can often only be drawn off from the rod by putting a stout pipe between the piston and the crosshead nut, and heating it. As it expands the piston is forced from the cone. To lift

the junk rings from the inside of large cylinders, which are accessible through manholes, tapped holes are sometimes made in the cylinder covers for eyebolts. This is especially necessary if, above the larger cylinder, there is a high-pressure cylinder, which would otherwise have to be dismantled every time any repairs had to be carried out in the lower cylinder.

§ 99. **Piston Rods.**—In large engines the crosshead is not, as a rule, forged in one piece with the piston rod; in small engines, however, and where it is necessary to economise weight, the two are forged in one. In auxiliary engines the piston and the rod are also sometimes forged in one piece.

Separate piston rods are usually fitted into the crosshead and piston with coned ends, and held in place by a nut. At the piston end the cone or taper is 1 in 5 to 1 in 7 when it is not necessary to take out the rod frequently (merchant ships, &c.); 1 in 3 to 1 in 4 when the rod has to be removed from the piston to examine the latter (engines below the armoured deck of an ironclad, top piston of tandem engines, &c., and where the piston rod and crosshead are in one). Under these conditions a taper of 1 in 3 to 1 in 4 is used at the crosshead end, because it is at this end that the piston rod is most frequently removed. Cylindrical ends are also sometimes met with; if they are very accurately fitted, the result is satisfactory.

The diameter of the rod itself is made slightly larger than the largest diameter of the tapered ends, that the latter may not be injured during erection or tearing down. At the largest end of the tapered part there is usually a small collar, against which the piston rests, and which prevents the boss from splitting, or the piston from eventually becoming loose, should it be strained owing to water in the cylinder. (See Fig. 184A.)

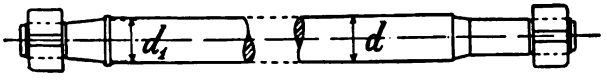


Fig. 184A.

The nuts of the piston rod are hexagonal, or are sometimes circular with two slots cut in them. A strong split pin, or a small set screw fitted against one side of each nut (see Figs. 180 and 172A) prevents it from slacking back. To prevent the piston turning on the rod, the larger end of the cone is generally fitted with a small key or set pin.

Dimensions of Piston Rods.—The diameter below the thread at either

end of the rod must be such that the maximum stress on the area at bottom of thread shall not exceed the following:—

5,500 to	7,000	lb. per square inch for	cargo boats.
7,000 „	8,500	„ „	mail steamers.
8,000 „	10,000	„ „	warships.
10,000 „	12,500	„ „	light cruisers and torpedo-boats.

The material used is assumed to be Siemens-Martin steel. In the rod itself the stress should only be about half the above figures, and, in this case, the breaking stress need not be taken into account. The diameter of the rod is about—

$d = d_1 + \frac{d_1}{10}$ for small rods.

$d = d_1 \times \frac{d_1}{15}$ for large rods, as an allowance for future turning down.

Tail Rods.—In large pistons the piston rod is frequently carried through the cover, in order that the piston may be well guided. The diameter of the tail rod is generally made slightly less than the diameter below the thread on the piston rod.

Piston rods are occasionally secured to the piston by a flange. In

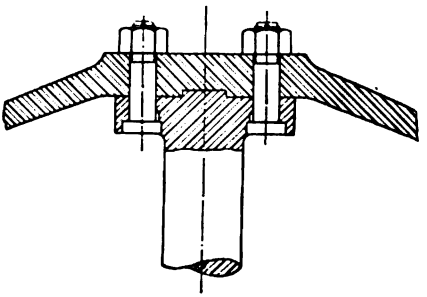


Fig. 184B.

such cases the coupling bolts must be very accurately fitted, and the nuts must be well locked. The advantage of this arrangement is the ease with which the piston can be taken off, against which must be set the difficulty of preventing the bolts from slacking back. (See Fig. 184B.)

Material.—The material used for the piston rod is almost exclusively Siemens-Martin steel, and sometimes crucible or nickel steel (for warships). It should be fairly hard, so that the rod does not wear quickly, or get ridged at the ends.

Connecting Rod and Crosshead.

§ 100. **The Length of the Connecting Rod** is almost always made as great as possible, so as to reduce the pressure on the guides to a minimum. The ratio of the length of the connecting rod to the

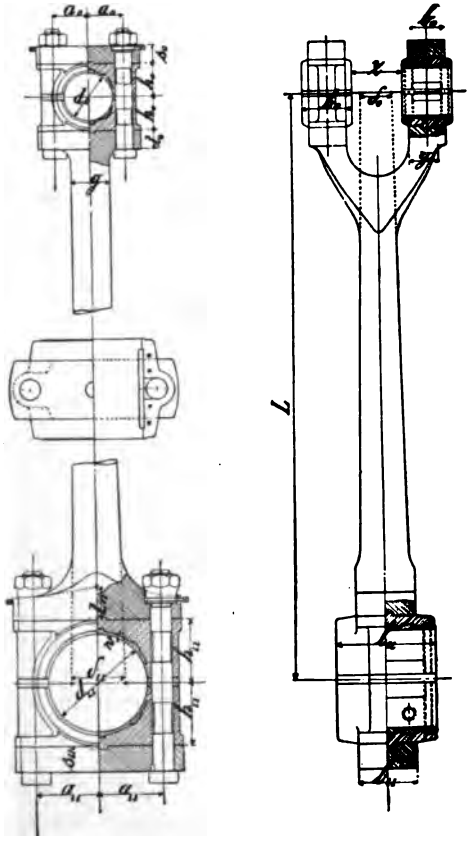


Fig. 185.

radius of the crank is scarcely ever less than 4 to 1; but the restricted height of marine engines seldom allows of a larger ratio than 4·5 to 1.

The most usual types are shown in Figs. 185 and 186. In engines for merchant vessels, and frequently in those for warships, the crosshead bearings are on the connecting rod, the pin being fixed to the crosshead itself. If the crosshead and piston rod are forged in one piece, the bearing is generally in the crosshead, and the pin forms the upper

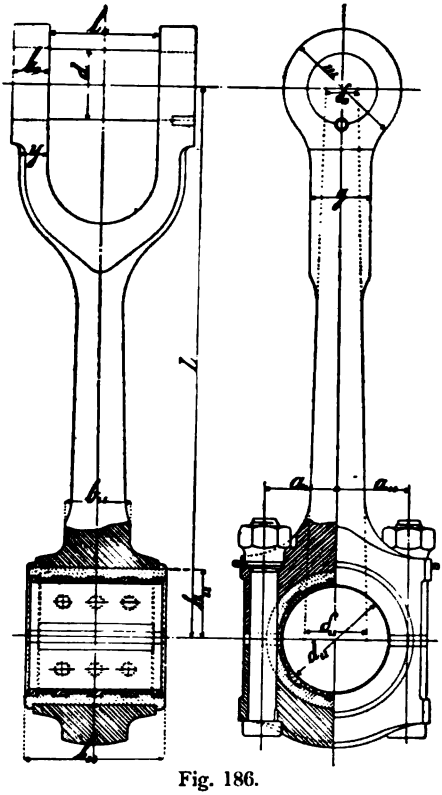


Fig. 186.

end of the rod. (See Fig. 186.) This latter arrangement is frequently used in warships, and in small engines for steam pinnaces, auxiliary engines, &c.

§ 101. **Connecting Rod.**—The diameter of the rod just below the fork is generally taken as equal to the diameter of the piston rod. The diameter of the shaft produced to the centre of the crosshead pin (Fig. 187) will therefore be δ_o = about 0·75 k, k being the diameter of the piston rod. The shaft is made thicker at the bottom end, firstly, to make the change of form from the large bottom end less sudden; and

secondly, to secure greater strength. Fairly correct values for the larger diameter δ_u are obtained if δ_u = about $0{\cdot}6d_u$. Connecting rods are occa-

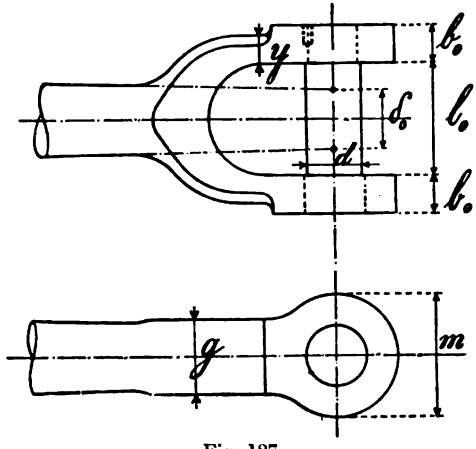


Fig. 187.

sionally made hollow, but this method of construction may prejudicially affect the solidity of the fork.

§ 102. **Connecting-rod Fork.**—This is generally the weakest and most dangerous part of the connecting rod, and its proportions should at least be calculated approximately as follows:—

The cross section at y (Figs. 187 and 188) has to take the greatest pressure and bending strain. The greatest pressure to which it is exposed will be—

$$s = \frac{P}{2} \times \frac{1}{y \cdot g} + \frac{\frac{P}{2} \times p}{\frac{y^2 \times g}{6}} = \frac{P}{2y \times g}\left(1 + \frac{6p}{y}\right)$$

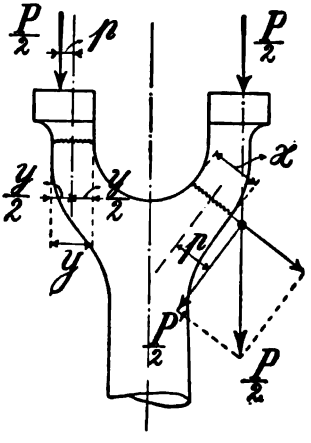


Fig. 188.

The first equation is the direct tensile or compressive stress produced by the maximum load $\frac{P}{2}$; the second equation gives the greatest tensile or compressive stress produced in the outer edges of the cross section y by the moment $\frac{P}{2} \times p$. The stress on any cross section x of the cross-

head fork may be calculated in the same way. The maximum load $\frac{P}{2}$ yields a component $\frac{P'}{2}$, which gives rise to a bending moment of $\frac{P'}{2} \times p$ about the section x, and at the same time sets up a tensile or compressive stress in this section. The greatest tensile or compressive stress upon the section x is thus—

$$s = \frac{P'}{2} \times \frac{1}{x \times g} + \frac{\frac{P'}{2} \times p}{\frac{x^2 \times g}{6}}$$

The shape of the fork must be determined by calculating the section at various points. If Siemens-Martin steel is used, and s = 5,500 to 11,000 lb. per square inch, the usual dimensions for the fork will suffice, the higher stresses being used for very lightly built engines. With small rods s is generally = 7,000 to 10,000 lb. per square inch where weight has not to be economised, but there are rods in use both in very large and small engines, in which the combined compressive and bending stresses are as much as 14,000 lb. per square inch.

Crosshead Pin.—Length of pin bearing = about $1\frac{1}{2}$ × its diameter. The dimensions may be determined from the pressures given in Table No. 23. The pin must of course also be sufficiently strong to resist bending stresses. It may be made hollow, if saving of weight is of importance. It is fitted into the fork by hydraulic pressure, or shrunk into it while the fork is hot. It is usual to key the pin in the fork by means of a strong set screw, as shown in Figs. 186 and 187. To fit the pin in more easily, it is made from $\frac{1}{16}$ to $\frac{1}{8}$ inch larger at one end, where it is held by the fork. The dimensions of the fork are as follows (Fig. 187):—

m = about $1{\cdot}8d$ to $2d$.
l_o = „ $1{\cdot}5d$ „ $2d$.
b_o = „ $0{\cdot}6d$.

The width g is slightly greater than the diameter of the shaft at the same place; the thickness y is calculated to resist the tensile, compressive, and bending stresses as given above.

§ 103. **Crank-pin Brasses.**—The dimensions of these brasses are determined by the maximum allowable pressure on the bearing surfaces. (See page 181.)

TABLE No. 23.

Maximum allowable Pressure on the Crank-pin and Crosshead Brasses.

Type of Vessel.	Crank-pin Brasses, lb. per square inch.	Crosshead Brasses, lb. per square inch.
Cargo boats	350 to 500	700 to 1000
Mail steamers, &c.	425 „ 570	850 „ 1000
Fast steamers, ironclads	570 „ 700	1000 „ 1250
Small cruisers, &c.	700 „ 850	1250 „ 1700
Torpedo-boats and destroyers	850 „1000	1700 „ 2100

The form shown in Fig. 185 is the one most commonly used for the connecting-rod head. For pins of large diameter the brasses are made of cast steel or bronze (the latter now only in warships) with white metal linings, and for small connecting rods they are often of gunmetal without any white metal, especially in the case of the crosshead bearings.

In Fig. 185, let

$$h = \frac{d}{2} \times 1{\cdot}3 \text{ to } h = \frac{d}{2} \times 1{\cdot}4$$

for cast-steel brasses lined with white metal, or for gunmetal brasses where no white metal is used; and

$$h = \frac{d}{2} \times 1{\cdot}35 \text{ to } h = \frac{d}{2} \times 1{\cdot}5$$

for gunmetal brasses lined with white metal, the higher values being taken for smaller, and the lower for larger pins.

The thickness of the white metal should be—

$w = \frac{11}{16}$ to $\frac{3}{4}$ inch if $d = 20$ to 24 inches.
$= \frac{5}{8}$ „ $\frac{11}{16}$ „ $= 15$ „ 20 „
$= \frac{9}{16}$ „ $\frac{5}{8}$ „ $= 12$ „ 15 „
$= \frac{1}{2}$ „ $\frac{9}{16}$ „ $= 8$ „ 12 „
$= \frac{3}{8}$ „ $\frac{1}{2}$ „ $= 6$ „ 8 „
$= \frac{1}{4}$ „ $\frac{3}{8}$ „ $= 3$ „ 6 „
$= \frac{3}{16}$ „ $\frac{1}{4}$ „ = less than 3 „

To prevent the loosening of the white metal, circumferential dove-tailed grooves are left in the blocks, and similar shallow recesses round each edge keep the white metal from working out. The inner surfaces of the blocks are tinned before the white metal is run in. Packing

pieces from $\frac{1}{8}$ to $1\frac{1}{8}$ inch thick, according to the diameter of the pin, are placed between the brasses, and very thin strips of metal from 25 to 35 S.W.G. thick are also inserted, for adjusting the brasses. The "play" allowed when fitting them is about $\frac{1}{1000}$ to $\frac{1}{800}$ the diameter of the pin. The crank-pin brasses are allowed a lateral play of from $\frac{1}{64}$ to $\frac{1}{8}$ inch on each side. In order to prevent the brasses from moving, they are secured in the head of the rod and in the cap by strong central pins, or (which is less desirable) by a slot parallel to the centre line of the pin (Fig. 189).

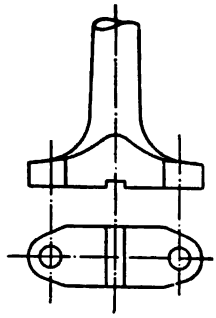


Fig. 189.

Circular-shaped Brasses.—These are often employed for the heads of connecting rods (see Fig. 186), and are rather lighter than the brasses described above, and shown in Fig. 185. If made of gunmetal their thickness should be as above, $h = \frac{d}{2}$. When round brasses are used, the packing strips between them must be very strong, to prevent the brasses from turning. If there are set pins for this purpose in the crown of the brasses, it is very difficult to fit them accurately into the connecting rod.

§ 104. **Connecting-rod Bolts.**—These are placed as close together as possible, and in order not to weaken the bolt a fine thread is generally used. (See Table No. 71, Part VII.) The nuts have the Penn locking device (Table No. 72, Part VII., page 608). For small rods lock nuts are used with split pins. To afford greater facility in fitting up and dismantling, the nuts are always placed at the top, and the head of the bolt at the bottom or cap end.

The maximum stress on the bolts at the bottom of the thread should be

5,000	to	7,000	lb. per square inch	for cargo steamers.
7,000	„	8,500	„ „	passenger ships and ironclads.
8,500	„	11,000	„ „	light war vessels.

A feather prevents the bolts from turning; those for the crank-pin end have tapped holes for eyebolts, so that they may be more easily removed. The crank-pin bolts are made a tight fit in the head of the rod and in the cap, but are given a little play in the brasses. The nuts are generally stamped with graduations, so that it may be seen at a glance exactly how far they have been screwed up.

Bearing Caps.—The usual dimensions for these can be arrived at if they are considered as a beam having an equally distributed load between

the two bolt centres. Let P be the maximum load, equally distributed, then the bending moment will be (Fig. 190)—

$$M = \frac{P \cdot 2a}{8} = \frac{P \cdot a}{4}$$

Modulus of cross section—

$$z = \frac{b \cdot s^2}{6} \cdot = \frac{P \cdot a}{4} \times \frac{1}{s}$$

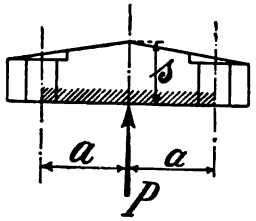


Fig. 190.

The safe stress is taken at s = 5,500 to 8,500 lb. per square inch, the higher value being adopted for light engines. Width of the cap (see Fig. 185) b_u = about δ_u for the bottom end, and $b_u = (1{\cdot}5 \text{ to } 1{\cdot}7) \times y$ for the top or fork end. If the crank pin be very long, b_u may be made wider, to avoid weakening the head of the rod.

§ 105. **Crosshead and Guide.**—In large engines for merchant vessels the crosshead consists, as a rule, of a square steel block with forged pins at each side. The guide blocks are generally made separate of cast iron or cast steel, and are secured in position by suitable bolts, the piston rod being connected by means of a cone and nut. In small engines, and those in which weight has to be economised, the piston rod, crosshead, and guide shoe are forged in one piece, and the latter faced with gunmetal or cast iron.

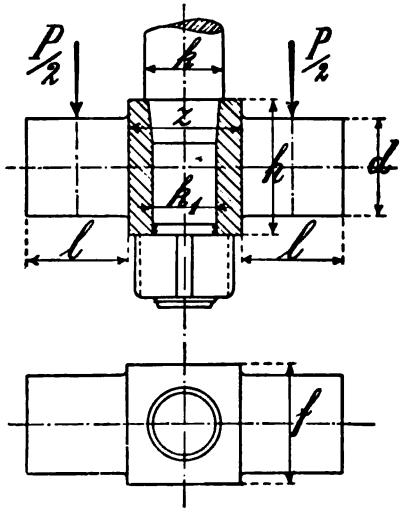


Fig. 191.

Where the crosshead and piston rod are not forged in one (Fig. 191), $z = 2 \times k_1$ = double the diameter of the screwed end of the piston rod over the threads. The height h and the width f are determined from the bending strain at the centre of the cross section. The bending moment at this point is—

$$\frac{P}{2} \times \left(\frac{l}{2} + \frac{z}{2}\right) = \frac{P(l+z)}{4}$$

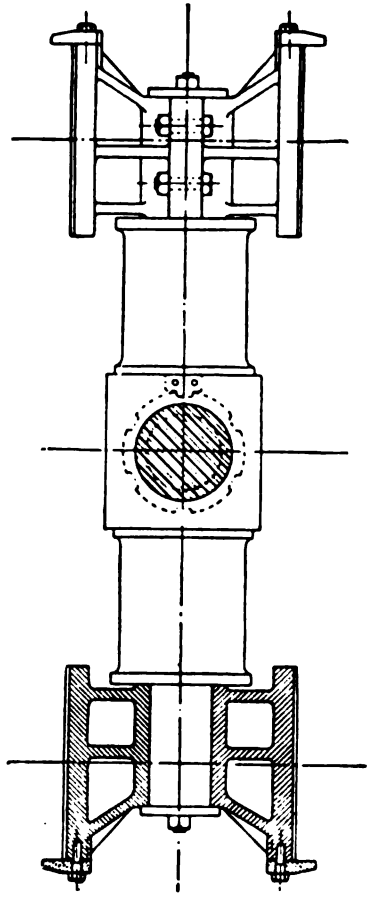


Fig. 193.

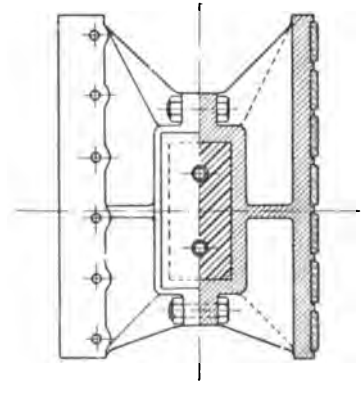


Fig. 192.

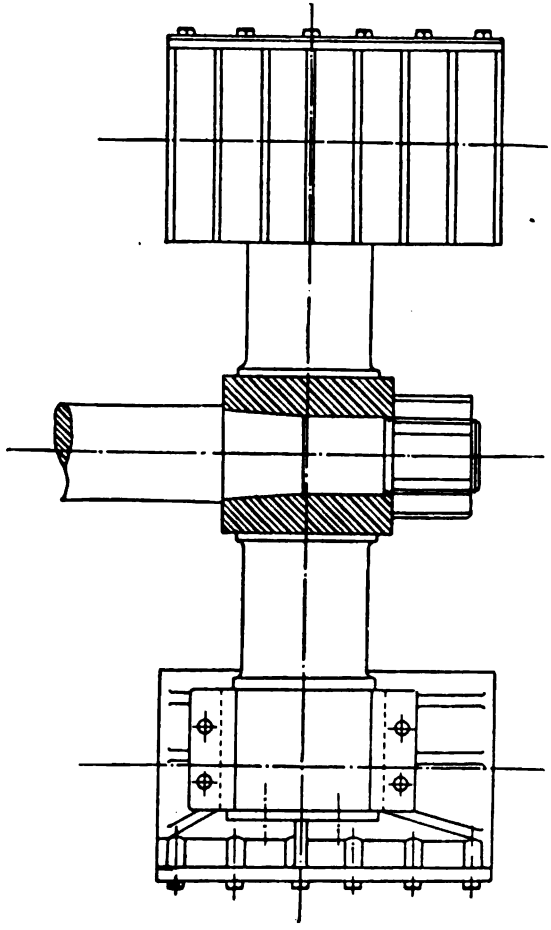


Fig. 195.

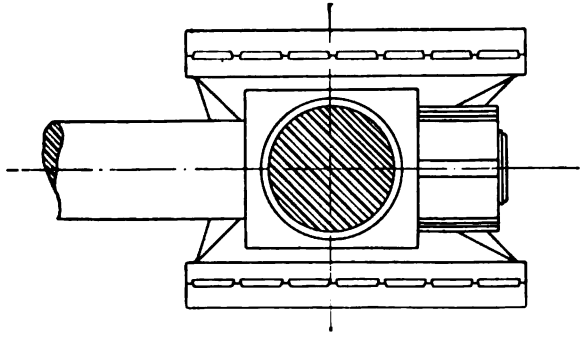


Fig. 194.

The maximum stress upon the central section will therefore be—

$$s = \frac{\frac{p(l+z)}{4}}{\frac{(f-k_1)h^2}{6}}$$

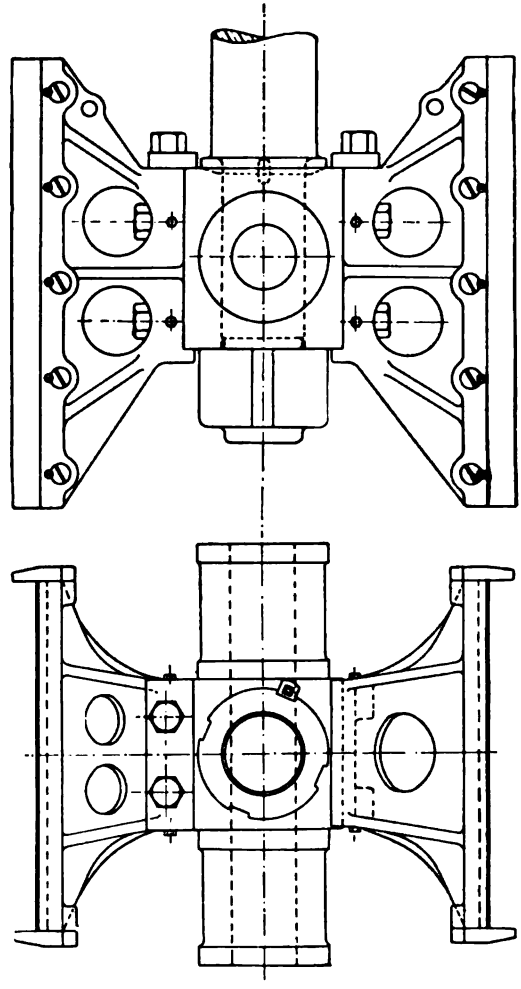


Fig. 196.

The value of s may be taken at 7,000 to 10,000 lb. per square inch, according to the type of engine.

As a rule f should not be made less than $1{\cdot}8 \times k_1$; the height h being

taken as large as possible. The length of the pin at each side is generally made equal to its diameter, *i.e.*, $l = d$. The pin has no fillet on the

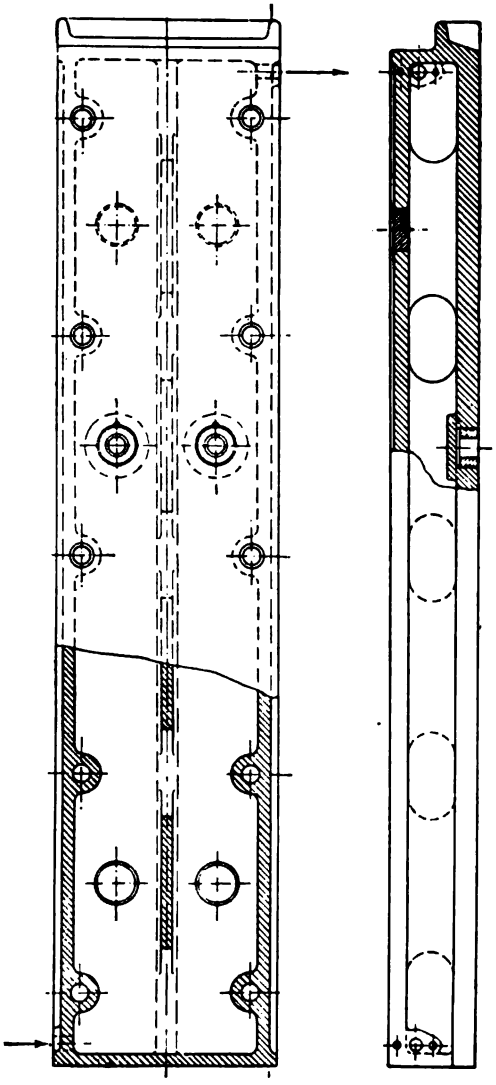


Fig. 197.

outer end, because as little play as possible is allowed between the crosshead bearings and the body of the crosshead. The pins are slightly flattened on the sides, so as to form a recess to retain the

lubricant, and thus ensure more efficient lubrication at the top and bottom of the pins. Very large engines having four guide columns are usually fitted with four slipper blocks. This is a good arrangement, as it facilitates inspection (Figs. 192 to 195).

Method of fitting the Slipper Blocks to Crosshead.—The shoes must be capable of being easily taken down, and yet be so well fitted that there is no possibility of their working loose. As the faces wear, the slippers must be capable of being easily adjusted. This is usually done by fitting thin strips of metal between the body of the crosshead and the slippers, or by providing a wedge-shaped piece between them, which can be drawn up. The arrangement shown in Fig. 196 is often used. The slippers are slipped in from below, and secured by plates and a few strong set screws. They are also held at the sides by set screws. The *slippers* are made of cast iron or cast steel. The bearing surfaces consist of flat strips of white metal held in position by slightly wedge-shaped grooves, or of plates of white metal cast on, and secured to the slipper by lateral dovetailed grooves. Of the two, the first arrangement is preferable. (See Fig. 192.) In large engines side plates are generally fixed to the slippers, to guide them laterally. These plates have the further advantage of preventing the white metal from being forced out. They are generally secured in position with countersunk set screws. (See Fig. 196.)

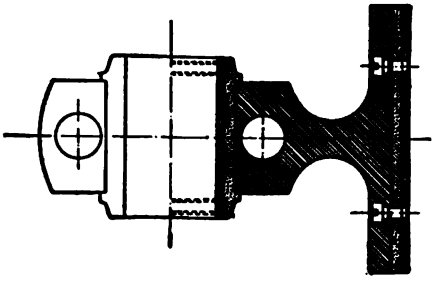


Fig. 198.

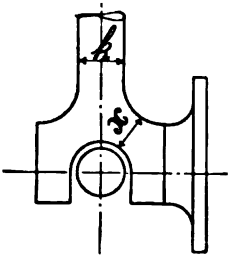


Fig. 199.

§ 106. **Crossheads Forged in one with the Rod.**—These are employed in nearly all small marine engines, and in large engines where weight has to be economised. The bearing is generally in the crosshead, and the pin is held in the forked end of the connecting rod. The crosshead has generally only one slipper guide on the ahead side, and plates overlapping the sides of the shoe take the thrust when going astern. The shoe and crosshead are generally forged in one piece, and

the slipper is made of cast iron or gunmetal, screwed on to the shoe with a shoulder above and below, so that the two faces can be accurately fitted together. As a rule the slippers are faced with white metal. The back of the shoe is often provided with fitting strips, to prevent any cutting of the guide plates when running astern. Figs. 198 to 200 show an arrangement of crosshead which is often used. The dimensions of the brasses and cap bolts are similar to those given for the connecting rod in § 104. The width at x (Fig. 199) is made as large as possible, in order that, when the inner brass is driven into position, it may not throw the piston rod out of line with the slipper face. The brasses are made of gunmetal lined with white metal, and both these are made the same thickness as for the connecting rod. (See § 100.)

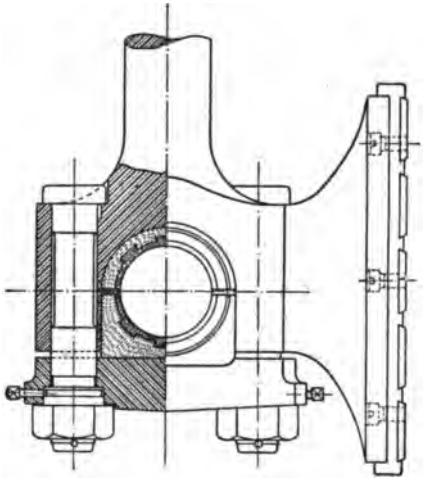


Fig. 200.

§ 107. **Pressure on the Guides.**—If $\frac{r}{l}$ is the ratio of the radius of the crank to the length of the connecting rod, the pressure on the guide, which is taken as a basis for calculating the dimensions of the latter, will be—

$$\text{Pressure on the guide} = \frac{r}{l} \times \text{maximum load.}$$

§ 108. **Guides.**—The surface of the guides may be determined from the following permissible pressures. That is—

55 to 65 lb. per square inch for cargo and slow-running passenger steamers.
65 to 80 lb. per square inch for mail steamers.
70 „ 85 „ „ ironclads and large cruisers.
85 „ 120 „ „ small cruisers and torpedo-boats.

The width of the guide face depends on the arrangement of the supporting columns. As a rule—

$$\frac{\text{Length}}{\text{Width}} \text{ of guide} = \frac{5}{4} \text{ to } \frac{5}{3}$$

Cast iron is the most suitable material for the guides; for this reason, in all engines which have no cast-iron columns, special cast-iron guides have to be fixed to the columns. It is only in small engines that the front of the column itself can be utilised as the guide. An engine can be more easily erected, when separate guides are provided.

In large engines the guides are almost always internally cooled with water, and are therefore made either as separate chambers (Figs. 197 to

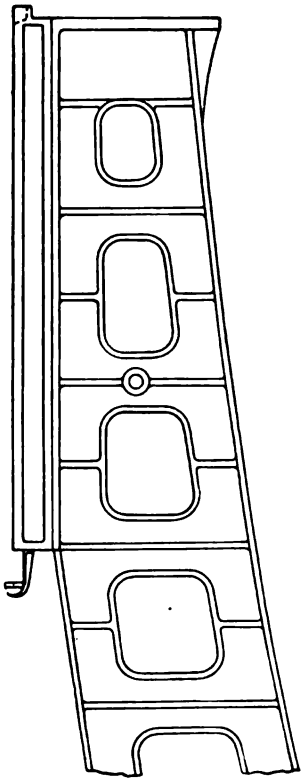


Fig. 201.

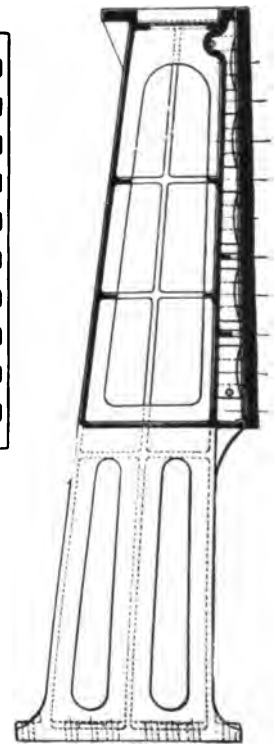


Fig. 202.

201), or as the front wall of a hollow recess in the column (Fig. 202). They are secured by bolts or set screws with countersunk heads. A few oil-ways are cut diagonally across the guides, and there is a lubricator at the top and an oil receiver at the bottom, out of which a comb on the slipper picks up the oil, and smears it over the guides. The water for cooling them is generally admitted at the bottom, and discharged at the top. The guides are often fitted in

such a way that the front and back faces, when cold, are rather nearer to each other at the top than at the bottom, to allow for expansion of the cylinder when hot. In order to hold the piston and crosshead in their highest position when the connecting rod is being taken down, two large tapped holes are made in the guide, and set screws fitted in to hold an iron plate which supports the crosshead. The arrangement of the guides usually adopted in engines entirely supported on steel or wrought-iron columns is shown in Figs. 203 and 204. In all engines

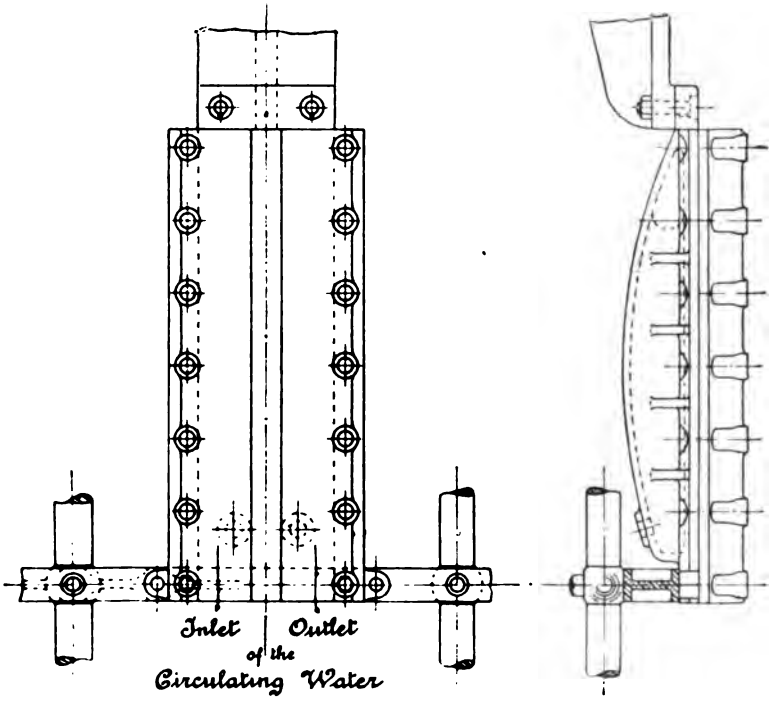


Fig. 203.

in which weight has to be minimised, back plates are fitted, instead of a separate astern guide. These back plates are of cast iron in small, and of steel or gunmetal in larger engines. Either the crosshead or the back plates are faced with bronze or white metal, so that steel does not run upon steel, as this is likely to cause abrasion.

The guide plates are always fitted so that they can be removed from either side as the strips wear, without having to take down the whole of the crosshead, piston, and rod. New gunmetal strips can then be fitted on to the slipper.

If $\frac{P}{2}$ is the pressure upon the working faces of the guide plates, the strain upon the bolts holding them will be $\frac{P}{2} \times \frac{a+b}{a}$ (see Fig. (204).

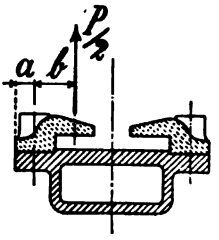


Fig. 204.

As the guide plates are not made to resist much bending strain, there should be as many set screws or bolts, per unit length of the slipper, throughout the length of the guide, as will prevent the permissible strain being exceeded. The method of fixing the guide plates to the guides must be such that, in taking down the plates, the guide itself is not completely disconnected from the framing of the engine. Some bolts in addition to those holding the plates to the guide should be provided, to secure the main guide to the framing.

Crank Shafts.

§ 109. **Crank Shafts.**—The turning moment of the crank, corresponding to the maximum load on the engine, is generally taken as the basis for calculating the torsional stresses in the crank shaft. If the crank shaft is built up of several sections, each section must be of the same strength as the end section, which has to take the total torsional stress.

The mean turning moment (M) of the shaft, in inch pounds, is—

$$M = \frac{\text{I.H.P.}}{n} \times 63{,}000$$

where I.H.P. is the indicated horse-power and n is number of revolutions of the engine. From this moment, the diameter (D) of the shaft, if *solid*, is calculated as follows:—

$$D^3 = \frac{16}{\pi} \times \frac{M}{S} \text{ and } D = 1{\cdot}72 \times \sqrt[3]{\frac{M}{S}}$$

and for *hollow* shafts—

$$D^3 = \frac{16}{\pi} \times \frac{M}{S}\left(\frac{1}{1-\frac{d^4}{D^4}}\right), \text{ and } D = 1{\cdot}72 \times \sqrt[3]{\frac{M}{S}\,\frac{1}{1-\frac{d^4}{D}}}$$

the diameter of the bore being $= d$.

The allowable torsional stress is usually as follows:—

s = 4,000 to 4,500 lb. per square inch in engines for cargo and passenger steamers.
s = 5,000 to 5,500 lb. per square inch in engines for heavy war vessels.
s = 5,500 „ 6,500 „ „ „ light „

These allowances for stress may appear small, but should not be exceeded, in view of the various and exceptional strains upon the crank shaft, such as bending, if the bearings do not wear uniformly, racing of the screw in a rough sea, &c. If the maximum turning moment (as shown in the tangential diagram of forces at page 63) is taken as the basis for calculation, the above stresses can be increased in the ratio of 1 to from 1·2 to 1·5, the higher figure being used for engines with one crank, and the lower for two, three, or more crank engines.

Hollow Shafts.—Shafts are made hollow to lighten them, and to distribute the material more efficiently. It is usual to take $\frac{d}{D}$ = about 0·4 to 0·6. After the shaft is erected, the mouth of the bore should be closed up, so that the inside of the shaft may not become rusty, and also to prevent water reaching the bore, and getting between the coupling flanges and corroding them.

§ 110. Lloyd's Rules for Determining the Sizes of Shafts (July 1902).

"RULE 60. The diameters of intermediate shafts are to be not less than those given by the following formula:—

For compound engines with two cranks at right angles, diameter of intermediate shaft in inches =

$$(\cdot 04_A + \cdot 006_D + \cdot 02s) \times \sqrt[3]{P}$$

For triple-expansion engines with three cranks at equal angles, diameter of intermediate shaft in inches =

$$(\cdot 038_A + \cdot 009_B + \cdot 002_D + \cdot 0165s) \times \sqrt[3]{P}$$

For quadruple-expansion engines with two cranks at right angles, diameter of intermediate shaft in inches =

$$(\cdot 034_A + \cdot 011_B + \cdot 004_C + \cdot 0014_D + \cdot 016s) \times \sqrt[3]{P}$$

For quadruple-expansion engines with three cranks, diameter of intermediate shaft in inches =

$$(\cdot 028_A + \cdot 014_B + \cdot 006_C + \cdot 0017_D + \cdot 015s) \times \sqrt[3]{P}$$

For quadruple-expansion engines with four cranks, diameter of intermediate shaft in inches =

$$(\cdot 033_A + \cdot 01_B + \cdot 004_C + \cdot 0013_D + \cdot 0155s) \times \sqrt[3]{P}$$

Where A = diameter of high-pressure cylinder in inches.
B = diameter of first intermediate cylinder in inches.
C = diameter of second intermediate cylinder in inches.
D = diameter of low-pressure cylinder in inches.
S = stroke of pistons in inches.
P = boiler pressure above atmosphere in pounds per square inch.

The diameter of crank shaft, and of thrust shaft under the collars, to be at least $\frac{21}{20}$ of that of the intermediate shaft. The diameter of thrust shaft may be tapered off at each end to the same size as that of the intermediate shaft.

The diameter of the screw shaft to be equal to the diameter of intermediate shaft (found as above) multiplied by $\left(\cdot 63 + \frac{\cdot 03 P}{T}\right)$, but in no case to be less than 1·07T, where P is the diameter of propeller shaft, and T the diameter of intermediate shaft, both in inches.

This size of screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in paragraph 33 below (rule for material). If no liners are used or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$ that given above.

The diameter of screw shaft is to be tapered off at the forward end to the size of the crank shaft.

Note.—These rules are intended to apply to two-cylinder compound engines in which the ratio of areas of low and high pressure cylinders does not exceed 4·5 to 1; to triple-expansion engines in which it does not exceed 9 to 1; to quadruple-expansion engines in which it does not exceed 12 to 1; and in all cases, as regards the stroke, in which the length of stroke is not less than one-half the diameter, or greater than the diameter of the low-pressure cylinder. Engines of extreme proportions beyond these limits must be specially submitted to be dealt with on their merits."

LLOYD'S RULES FOR MATERIALS OF SHAFTS (July 1902).

"RULE 31. *Shafts.*—All shafts are to be made of good material and are to be examined when rough turned and when finished. In the case of screw shafts, scrap steel is not to be used. It is recommended that these be made of ingot steel, or forged from blooms made from rolled iron bar of good fibrous quality.

A tensile and a bend test are to be made on pieces cut from one end of each ingot steel shaft forging, the piece from which they are cut being of the same size as the body of the forging. In the case of built crank shafts, the tests are to be taken from the material of the crank pins and journals, not from the webs. If more than one piece is forged from one ingot, one test only will be required from the ingot. The tensile strength is not to be less than 27 tons per square inch, nor to exceed 32 tons per square inch. The elongation is not to be less than 30 per cent. in a length of 2 inches, measured on a plain portion turned not more than $\frac{3}{4}$ inch diameter. The bend test piece is to be made 1 inch square and must be capable of being bent cold, without fracture, through an angle of 180° over a radius not greater than $\frac{1}{2}$ inch." Cast steel is practically never used for crank shafts, not even for the cheeks.

"RULE 32. Gauges of an approved description for testing the truth

of the crank shafts are to be supplied with all new engines, and adjusted in the presence of the Surveyor."

"RULE 33. The length of the stern bush is to be at least four diameters of the shaft. It is recommended that the shaft liner should be continuous the whole length of the stern tube, and that the after end should be tapered in thickness, and made watertight in the propeller boss. If the liner is made in two pieces, the joint should be burned. If the liner does not fit tightly at the part between the bearings in the stern tube, the space between the shaft and the liner should be charged or 'forced' with a plastic material insoluble in water, and non-corrosive. If two liners are used, it is recommended that they be tapered in thickness at the ends, and that the shaft should be lapped or protected between the liners. In this case, and also if no liners are used, the diameter of the shaft should be $\frac{1}{10}$ of that required for a shaft with a continuous liner."

§ 110A. Board of Trade Rules for Shafts.

"RULE 144. *Size of Shafting.*—Main, tunnel, propeller, and paddle shafts should not be passed, if less in diameter than that found by the following formulæ, without previously submitting the whole case to the Board of Trade for their consideration. It will be found that first-class makers generally put in larger shafts than those obtained by the formulæ.

For compound condensing engines with two or more cylinders, when the cranks are not overhung—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}$$

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right)$$

Where S = diameter of shaft in inches.

d^2 = square of diameter of high-pressure cylinder in inches, or sum of squares of diameters when there are two or more high-pressure cylinders.

D^2 = square of diameter of low-pressure cylinder in inches, or sum of squares of diameters when there are two or more low-pressure cylinders.

P = *absolute* pressure in pounds per square inch—that is, boiler pressure + 15 lb.

C = length of crank in inches.

f = constant from following table (Table No. 23A).

For ordinary condensing engines with one, two, or more cylinders, when the cranks are not overhung—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{3 \times f}}$$

$$P = \frac{3 \times f \times S^3}{C \times D^2}$$

Where S = diameter of shaft in inches.

D^2 = square of diameter of cylinder in inches, or sum of squares of diameters when there are two or more cylinders.

P = *absolute* pressure in pounds per square inch—that is, boiler pressure + 15 lb.

C = length of crank in inches.

f = constant from following table.

TABLE No. 23A.

For Two Cranks, Angle between Cranks.*	For Crank, Thrust, and Propeller Shafts.†	For Tunnel Shaft.
90°	1047 ‡	1221
100°	966	1128
110°	904	1055
120°	855	997
130°	817	953
140°	788	919
150°	766	894
160°	751	877
170°	743	867
180°	740	864
For three cranks, 120°	1110	1295

* When there is only one crank the constants applicable are those in the table opposite 180°.

† The constants in this column should be reduced by 15 % when dealing with propeller shafts of new vessels. The portion of the propeller shaft which is forward of the stern gland, and all the thrust shaft, with the exception of the part enclosed in the thrust bearing, may be of the same diameter as the intermediate tunnel shafting.

‡ For paddle engines of the direct-acting type, multiply constant in this column suitable for angle of cranks by 1·4."

§ 111. **Crank Pin.**—This is usually made of the same diameter as the crank shaft; or sometimes rather larger, in view of the pressure on its surface. The length of the crank pin is determined from the pressures given for "connecting rods" on page 197. Long crank pins should, if possible, be avoided, as they diminish the rigidity of the crank shaft. In the first place, the bending stress on the crank pin and crank shaft is greater (Fig. 205); and, in the second place, the torsional stress on the after crank cheek A (Fig. 206), which is carried round with the shaft and the forward cheek B, is increased. In very quick running engines (for torpedo-boats) the pins are generally very long; for, although the force of the piston working on the rod is great, the diameter of the shaft, on account of the high speed, is relatively small. The length of the pins can be reduced if they are made somewhat thicker than the shaft, and weight can be saved by making them hollow. The ratio of the length to the diameter of the crank pin is—

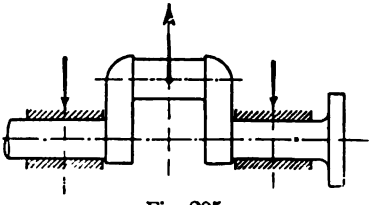


Fig. 205.

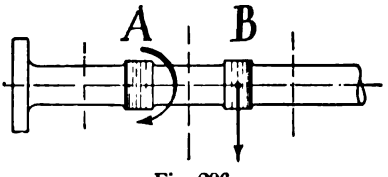


Fig. 206.

$\frac{l}{d}$ = 1·2 to 0·9 in engines for merchant vessels.

1·3	„ 1·1	„	heavy war vessels.
1·4	„ 1·2	„	light cruisers.
1·6	„ 1·4	„	torpedo-boats.

§ 112. **Built-up Crank Shafts** (see Figs. 207, 211, and 212).—These are used in all cases where weight is not the first consideration, and therefore nearly always in merchant vessels, but they are seldom employed in warships. Width of the cheeks (Fig. 207), $D = 1{\cdot}9$ to $2d$ (occasionally $D = 1{\cdot}8d$); and thickness of cheeks, $w = 0{\cdot}6$ to $0{\cdot}7d$. The end of the shaft, where it fits into the cheek, is generally strengthened

by making the diameter d_1, so that space may be left for a small fillet where the shaft rests in the bearing, and a gradual transition made between the cheek and the shaft—

$$d_1 = \text{about } d + \frac{d}{40}$$

The distance (z) between the crank pin and the shaft should not be

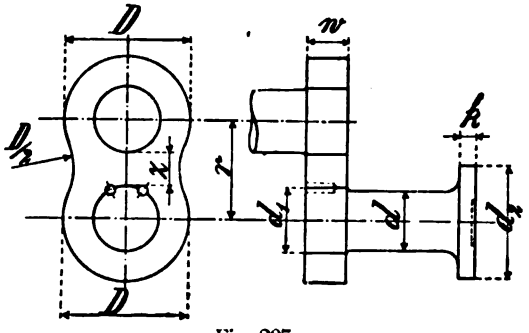


Fig. 207.

less than 0·45 to 0·50d, otherwise the pin and the shaft might wear loose. The stresses produced in the material round the pins by driving them in are doubled in the section $w \times z$, so that if z is too small, the material between the pins may be unduly stretched. The ends of the crank pin, which are shrunk into the cheeks, are left of the same diameter, or only very slightly thickened. The crank pin and shaft are fitted into

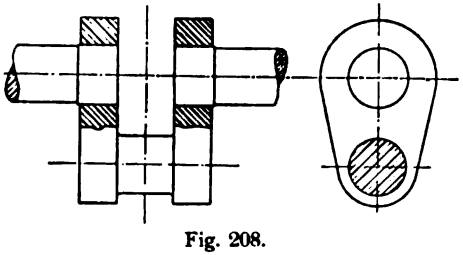


Fig. 208.

the cheeks while the latter are hot; the cheeks are therefore bored out previously to a diameter of $d_1 - \frac{d_1}{1,000}$. Sometimes the pin and shaft are forced into the cheeks by hydraulic pressure, and in this case also, the pin, before being driven in, is made $\frac{d_1}{1,000}$ larger than the hole. The shaft ends are keyed to the cheeks by flat keys or round pins; it is better to

have two small rather than one large key or pin, and to put them off the centre line (Fig. 207), so as not to weaken the section between the crank pin and shaft. They are driven into the cheeks to a depth of about $0{\cdot}7d$; the diameter, if pins are used and there is only one, is $= \frac{d}{15} + 0{\cdot}39$ inch for shafts of 4 inches diameter and upwards. The crank pins are sometimes keyed in a similar manner, but it is unnecessary.

In some crank shafts the crank pin and cheeks are forged in one piece, and only the shaft ends are fitted in. The disadvantage of this arrangement is the difficulty of securing a sound forging and of handling it, but with it a saving in weight and space is obtained. It is unusual to have crank shafts in which the coupling flanges are let into the cheeks (Fig. 209). The space for the flanges can only be spared at the cost

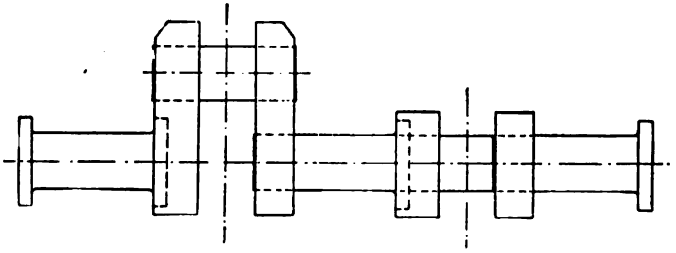


Fig. 209.

of considerably weakening the cheeks. At the most, this arrangement is permissible for the front crank and the one next to it; because the torsional stress upon them is less than that upon the after crank.

§ 113. **Crank Shafts with Crank Pin and Web forged in one Piece.**—These are considerably lighter than built-up crank shafts, but more difficult to manufacture, and are at the same time less solid, because the complicated shape of this class of forging is not conducive to strength. On account of its lightness, this kind of shaft is exclusively used in engines for war vessels, yachts, steam pinnaces, and other lightly-built engines. In torpedo-boats and ships' launches, where the weight has to be reduced to a minimum, crank shafts are sometimes forged with all the cranks and eccentrics in one piece; but the cost of constructing shafts in this way is enormous, and the manufacturing risks very great. The strength of the crank pin is the same as that given above; the thickness of the cheeks is also the same, viz., $w = 0{\cdot}6$ to $0{\cdot}7d$. Width of the cheeks, $b =$ about $d + 2\left(\cdot 2 + \frac{d}{40}\right)$. This gives the

depth of the fillet a = about $\cdot 2 + \frac{d}{40}$. The cheeks are generally rounded off top and bottom, while in the lathe, with centres M or C (Fig. 210). The outer sides are frequently tapered off as far as the centre of the crank pin, or of the shaft, but care must be taken not to cut off too much, and thus weaken the connection between the crank pin or shaft and the cheek. In the forged crank shafts of warships, both shaft and crank pins are nearly always made hollow.

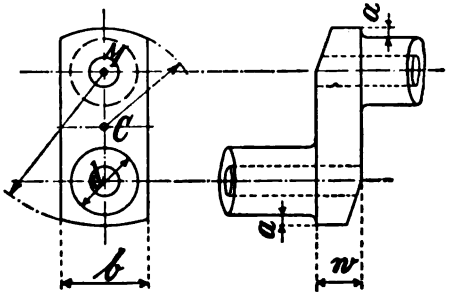


Fig. 210.

§ 114. **General Remarks on Crank Shafts.**—As a rule in merchant vessels separate lengths of crank shaft are made for each cylinder, and bolted together by means of flange couplings. In warships the whole crank shaft, consisting of two or three cranks, is often forged in one piece. If the engine has four cranks, they are forged two and two together. To forge two cranks in one piece has no advantage, beyond the saving of weight and space due to the absence of the couplings. The spare shaft is dearer and more complicated to make, more cumbersome and difficult to stow away. The separate sections of the crank shaft in the engines of merchant vessels should all be alike and interchangeable, necessitating only one spare piece of shafting. Even if the cylinders are at varying distances from each other, this arrangement can easily be carried out, by having one long and one short length of shaft in each crank section. (See Fig. 214 for a three-crank engine.)

For *angle of the cranks* refer to "Balancing the Moving Parts" (§ 37), and "Turning Moment of the Multiple-crank Engine" (§ 30). As a rule the cranks, with two-crank engines, are set at an angle of 90°, with three-crank engines at 120°, and with four-crank engines at 90°, or they are disposed in accordance with the Schlick system of balancing. (See "Arrangement of Main Engines," page 106.)

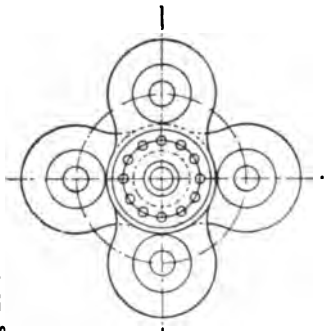


Fig. 211.

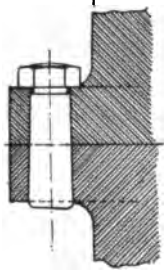


Fig. 213.

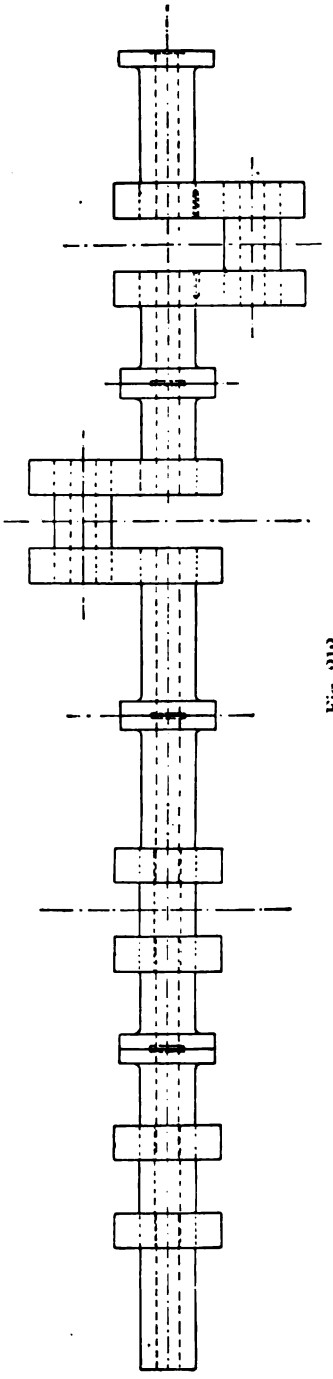


Fig. 212.

§ 115. **Crank-shaft Couplings.**—The various sections of shafting are bolted together by means of flange couplings. In warships sometimes easily disconnected couplings are fitted between the aft crank shaft and the tunnel shaft (see below), but these are now seldom used. The diameter of the flanges depends on the number and size of the bolts. Thickness of the flange $k = 0{\cdot}25$ to $0{\cdot}28d$. (See Fig. 207.) The coupling bolts are

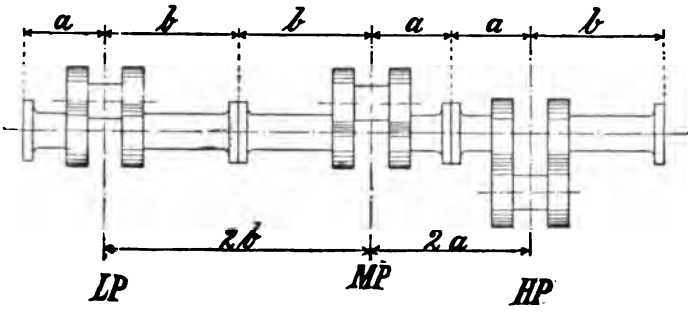


Fig. 214.

generally turned with a taper of 1 in 15 to 1 in 25. The mean diameter is used in calculating the shearing stress. Mean turning moment of shaft (see page 208)—

$$M = \frac{\text{I.H.P.}}{n} \cdot 63{,}000. \text{ in. lb.}$$

If r is the radius of the bolt circle, the shearing stress will be—

$$P = \frac{M}{r}$$

and the area corresponding to the mean diameters of the bolts—

$$F = \frac{M}{r} \times \frac{1}{S}$$

The stress s is assumed to be the same as the torsional stress on the crank shaft. (See above.) If a large number of small bolts are used, the flanges may be smaller, and this is the usual arrangement in warships. If there is a small number of larger bolts, larger flanges must be employed, and this is the general practice for merchant vessels. As a rule the diameter of the bolts in the middle of the taper = about $\frac{d}{4}$ to $\frac{d}{6}$. The bolts are tightened up by means of nuts; the diameter over the thread being made a good deal smaller than the diameter at the end of the taper, in order that the nuts may not be too

large. If thin nuts are used, they are usually fitted with a locking device and split pin. The holes for the coupling bolts must not only be accurately bored, but also carefully rimered out after both flanges have been fitted together. *It should be possible to draw the bolts from the couplings without having to take down any other part of the engine*, but it is often impossible to avoid removing the cap of the nearest bearing. The coupling flange runs into the shaft with a curve, the radius of which is about $\frac{d}{10}$; the bolts are set so closely to the body of the shaft that clearance for the nuts has generally to be made by recessing into the root of the flange. Concentric recesses are often turned in the face of the flanges, for inserting centring discs.

§ 116. **Materials for Crank Shafts** (compare Lloyd's Rules, page 210).

1. Wrought iron is used only for small, cheap cargo steamers.

2. Siemens-Martin steel, tensile strength 26 to 32 tons per square inch, with not less than 20 % elongation in 8 ins., for warships and merchant steamers.

3. Nickel steel, tensile strength 35 to 40 tons per square inch, with not less than 20 % elongation in 8 ins., for fast merchant steamers, iron-clads, &c.

4. Crucible steel for torpedo boats and destroyers, where the greatest care in construction is required. Tensile strength, 28 to 32 tons per square inch, and 20 to 25 % elongation in 8 ins.

Valve Gear Rods.

§ 117. **Power required to Drive the Valves.**—It is difficult to obtain accurate knowledge as to the power required to drive the valves, and the strain that the rods will have to stand. In flat slide valves the strain depends upon the coefficient of friction, and the varying steam pressures. In piston valves it depends on these factors, on the pressure exerted by the packing rings of the valves, and on the fitting of the valves. In multiple cylinder engines each valve gear is calculated to transmit the power required to work that slide valve which has the greatest load on it, the dimensions of the valve rods of the other cylinders being made the same.

The pressure p (above atmosphere) upon the valve is obtained from the following table (p_1 being equal to absolute boiler pressure):—

Type of Engine.	Compound.		Triple.			Quadruple.			
	H.P.	L.P.	H.P.	M.P.	L.P.	H.P.	M.P.$_1$	M.P.$_2$	L.P.
Merchant vessels and large war vessels	p_1	$0{\cdot}3p_1$	p_1	$0{\cdot}4p_1$ to $0{\cdot}5p_1$	$0{\cdot}12p_1$ to $0{\cdot}2p_1$	p_1	$0{\cdot}45p_1$	$0{\cdot}16p_1$	$0{\cdot}08p_1$
Small war vessels and torpedo-boats	...	...	p_1	$0{\cdot}55p_1$	$0{\cdot}25p_1$	...	...	...	...

The coefficient of friction should be taken at—

$f = 0{\cdot}15$ for the L.P. cylinder.
$f = 0{\cdot}2$,, M.P. and H.P. cylinders.

The total area of the back of the valve must be taken as the area on which the calculation is made, without taking into account any balancing arrangement which may be fitted, as the rods must be strong enough to work the valve, should the balancing arrangements for any cause fail to act. To calculate the strength of the rods a maximum load Q is taken, which is arrived at as follows:—

$$Q = f \,.\, L \,.\, B \,.\, p,$$

L and B being the overall length and breadth of the valve, f and p being respectively the coefficient of friction and the effective pressure as given above.

Example.—Let the slide valve of the M.P. cylinder of a triple expansion engine be 50 inches wide and 40 inches long. Boiler pressure (absolute) = 150 lb. per square inch. According to the above table $p = 0{\cdot}4p_1 = 60$ lb. per square inch. Therefore—

$$Q = 0{\cdot}2 \times 40 \times 50 \times 60 = 24{,}000 \text{ lb.}$$

If an engine is fitted with *piston valves only*, the dimensions of the rods are calculated from the diameter at the bottom of the thread of the actual valve rod, which can only be arrived at empirically. This diameter d for piston valves is $d = C \times D$, D being the diameter of the valve, and

$C = 0{\cdot}11$ to $0{\cdot}20$ in H.P. cylinders—115 to 215 lb. per square inch pressure (absolute).
$C = 0{\cdot}10$ to $0{\cdot}17$ in M.P. cylinders—45 to 115 lb. per square inch pressure (absolute).
$C = 0{\cdot}07$ to $0{\cdot}011$ in L.P. cylinders—15 to 45 lb. per square inch pressure (absolute).

The higher values of c here given are for higher pressures, *small* valves, and big heavy engines; the lower values for lower pressures, *large valves* (27 to 48 inches diameter), and light engines. The load Q, used to calculate the remaining parts of the gear for the particular type of engine in question, is obtained by multiplying the allowable stress on the area at the bottom of the thread of the valve rod by this area. (See below.)

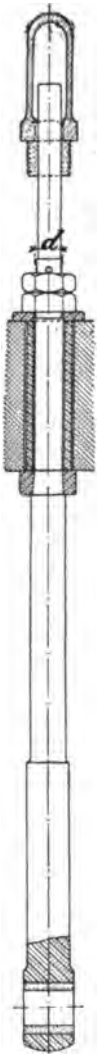


Fig. 215.

§ 118. **Valve Rods.**—The valve is supported at the bottom by a strong collar, and at the top by lock nuts. Between these two it should be able to work horizontally to and fro, so that the pressure of the steam may, if it is a flat D valve, hold it truly against the valve face; or so that the valve, if a piston valve, may work true in the liner. In order to get this adjustment the upper nut is fitted with a lock nut and pin, or a distance piece, composed of a piece of pipe, is fitted over the rod (Fig. 215).

It should be possible to draw out the rod from below through the stuffing box, when the rod and tail are forged in one piece, or when it cannot be drawn out from above, as in the case of a ship with an armoured deck. Instead of the collar mentioned above, therefore, a ring is often used, fitting tightly on to a cone on the rod (Fig. 215). As, when setting the valves, this ring may have to be turned down, to lower the valve, it is advisable to fit a flat ring above it. The conical ring can then remain intact, and any adjustment made on the flat ring above it. Care must be taken that the upper surface of the coned ring, as well as the collar of the valve bearing on it, are exactly at right angles to the axis of the valve rod, otherwise the valve will leak and wear unevenly. The *allowable stress in the valve rod at the bottom of the thread* (diameter at bottom of thread $= d$) produced by the load Q is—

s = 1,700 to 3,500 lb. per square inch for merchant vessels.
s = 4,500 to 5,500 lb. „ „ warships.

The lower portion of the rod, which works in the stuffing box and guide, is made very thick, partly on account of the coned ring just described, partly to facilitate the withdrawal of the rod.

Diameter of rod in the stuffing box is 1·7 to 1·9d.
„ „ guide is 1·8 „ 2·0d.

The taper in the ring below the valve is about 1 in 6.

In *small engines*, in which the eccentric rod is connected directly to the valve rod, the *head* of the latter consists of a simple eye bushed with gunmetal, which forms a bearing for the pin in the forked end of the eccentric rod. (See Fig. 215.) This bush is sometimes made in two parts, which can be adjusted by means of a cotter.

In *large engines*, and where the head of the valve rod works directly on the link block, it is similar in shape to the head of a connecting rod. The brasses are generally of gunmetal, round or polygonal outside, with strips of metal between, and as a rule without any white metal lining. Generally the head is forged in one with the rod, but it is sometimes fixed to it by means of a tapered sleeve and cotter. The allowable pressure on the effective surface of the bearing is $p = 570$ to 1,000 lb. per square inch. With link motion the width of the bearing is reduced as much as possible, in order that the fork of the eccentric rod may not have too large a gap. The valve rod should have a strong *guide*

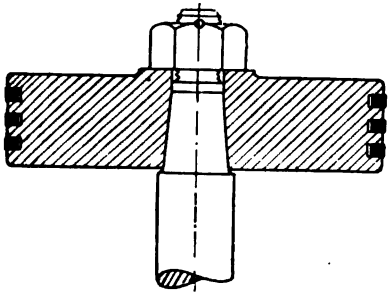


Fig. 216.

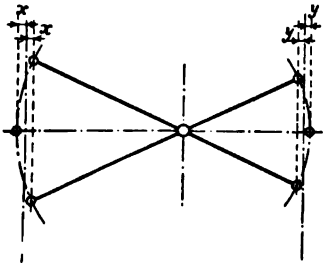


Fig. 217.

close to the valve chest, and as near as possible to the link and its reversing lever, &c. This guide is generally a simple casting of iron or steel with gunmetal bearings, and is sometimes made in two parts. This latter mode of construction is safer, and enables the rod to be more easily taken down at sea. At the top end, the valve rod generally works in and is guided by a gunmetal hood (Fig. 215).

Balance Cylinder Pistons (see Figs. 153, 154).—A balance piston working in an auxiliary cylinder often acts as the upper valve rod guide. The diameter of the piston is such that, at the mean working pressure in the casing, it just supports the weight of the valve. By this means the strain on the valve rod is nominally the same for the up and down strokes. The space above the piston is connected to the condenser. It is usual to fit balance cylinders in connection with very heavy piston or flat slide valves. The pistons of these balance cylinders are solid cast-iron discs, fitted either with two or three small steel or cast-iron Ramsbottom rings, or with one broad packing ring (Fig. 216).

To ensure smooth and quiet working of the valves, and to diminish the strain on the rods, "*Joy's*" *assistant cylinder* is sometimes used. Like the balance cylinder, it is fitted to the top of the valve chest. Shortly before the valve, and with it the balance piston, have reached their upper or lower extreme positions respectively, fresh steam from the main steam pipe is admitted above or below the balance piston. This forms a cushion, while the strain in the valve rods is reversed, and helps to work the valve after the change of stroke. The exhaust steam from the assistant cylinder is led into the valve chest to

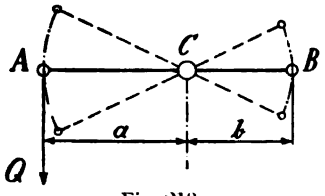


Fig. 218.

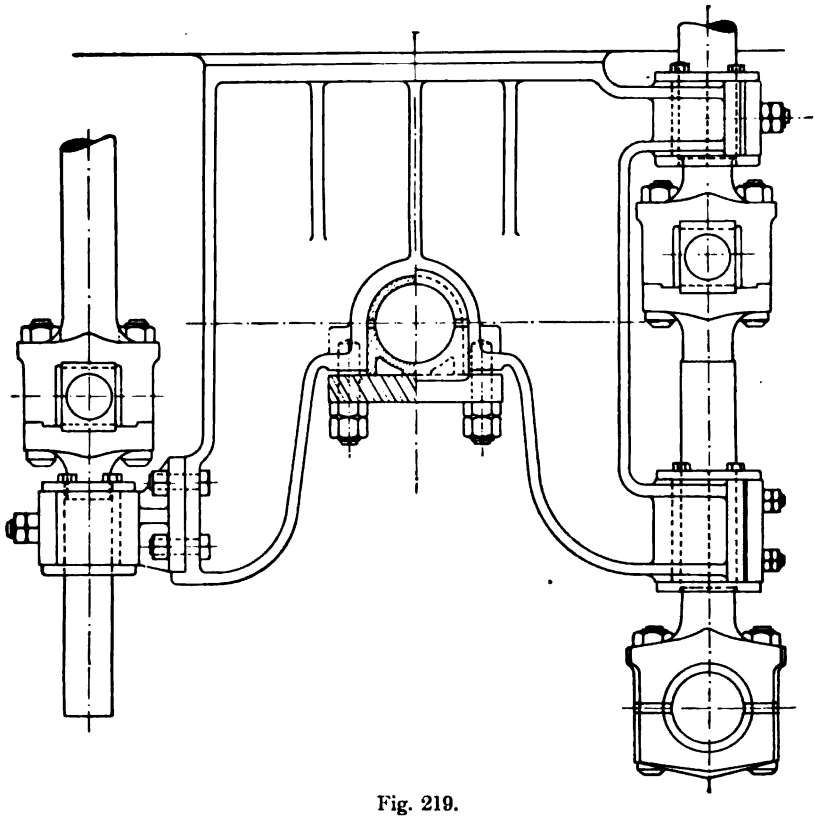


Fig. 219.

which it is fitted. To balance the weight of the valve, the steam is admitted sooner to, and discharged later from, the under than from the

upper side of the piston. The assistant cylinder is fitted with a guide bush having turned circumferential grooves, made as good a fit as

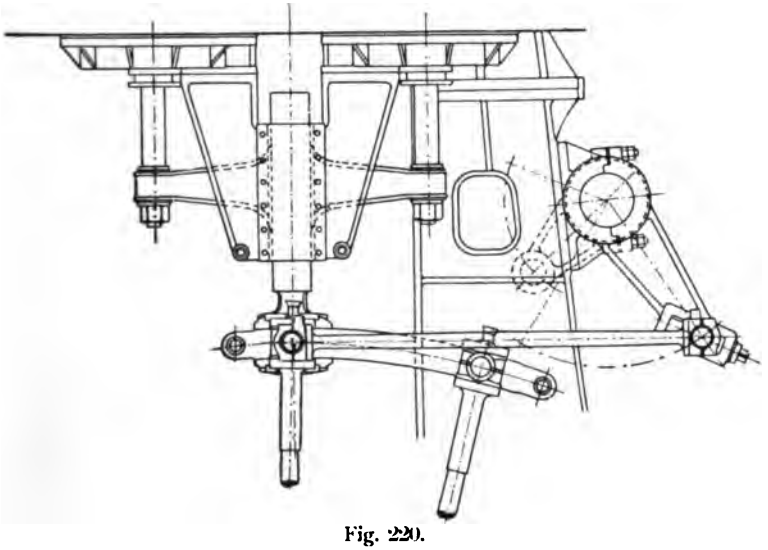


Fig. 220.

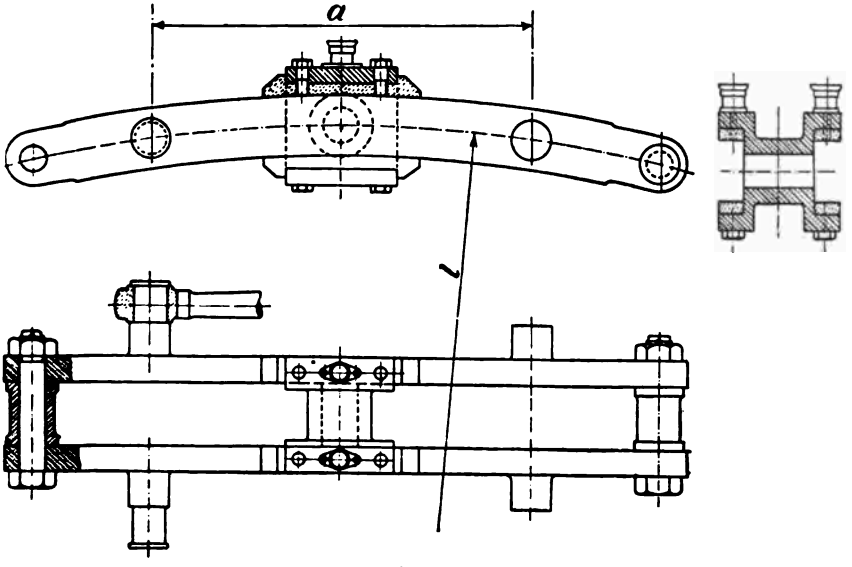


Fig. 221.

possible on the rod, so as to prevent any steam leaking from the cylinder into the casing.

§ 119. **Diameter of Rods.**—In valves which may have to be worked by hand, such as the reversing valves of steering engines, the valves of reversing gear engines, &c., care must be taken (compare Fig. 222) that the diameter of the guide rod is the same as that of

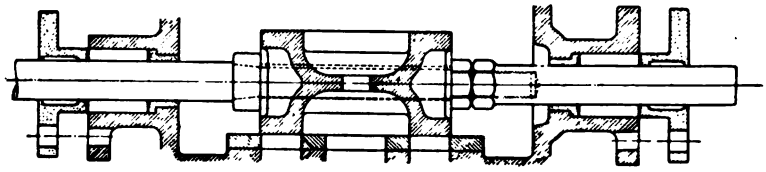


Fig. 222.

the valve rod. Otherwise the steam pressure on the difference of the area between d_1 and d_2 acts in the direction of the valve rod, and this may make any valves that have to be worked by hand difficult to move.

§ 120. **Valve Levers and Quadrants.**—It is often necessary to alter the line of action of the motion for working a valve, either parallel to itself or through a right or an acute angle from the longitudinal vertical axis of the engine; and this is done by means of two-armed levers of cast or forged steel. If such levers are used, the following points must be noted:—

1. The valve rods and levers must be so adjusted that the mean position of each arm of the lever is perpendicular to the line of motion of the valve or rod (Fig. 217).

2. The bearing block in the head of the rod, if so fitted, must slide freely, to allow for the swing of the lever (Fig. 218).

3. The valve rod should be well guided, and, as near as possible to the point where the lever joins it, there should be a small crosshead with a guide shoe, or simply a bushed guide block. (See Fig. 219.)

4. If the motion is transmitted by levers from the reversing link to the valve rod through an acute angle to the vertical axis of the engine, care must be taken that the centres of all the pins lie in one plane, in order to avoid any twisting strains upon the gear.

5. The pin A (Fig. 218), on which the valve rod works, is subjected to a load Q; the pin B, to a load $Q \times \frac{a}{b}$; and pin C, to a load $Q\left(1+\frac{a}{b}\right)$. It frequently happens that two piston valves or two slide valves are worked from the same reversing link, or that there are two valve rods for one slide valve. In such cases the rods are connected by a so-called

"cross" beam. This should be most carefully guided, and a little play given in the eye round each rod, to allow for expansion in the cylinder, and also for any little irregularities in the construction. The material for the crossbeam may be either wrought iron, steel, or cast steel. (See Fig. 220.)

§ 121. **Stephenson's Link Motion** (Fig. 221).—This consists, in principle, of two curved links, the mean radius of which = length of the eccentric rod. Both the links are of rectangular section, and their ends are held apart by distance pieces; the link block slides between them, and the pins for the eccentric and reversing rods are placed outside the bars.

Material Used.—This may be cast steel, or better still, forged steel.

Distance between centres of eccentric rod pins, $a = 5$ to $6 \times$ amount of eccentricity.

The length of the rubbing surfaces of the quadrant and the space between the two distance pieces are made to correspond with the full travel of the quadrant, from the full ahead to the full astern position. The travel is arrived at by drawing out the path described by the blocks at the extreme position of the links. The pins for attaching the reversing links are generally fixed on the quadrants at the "ahead" position (Fig. 221) in mercantile vessels; but in warships the point of attachment is frequently at the centre of the quadrants, so that the steam may be evenly distributed ahead or astern. The exact position on the quadrants of the point of attachment of the reversing lever is often dependent upon the actual design of the levers.

The link block consists of a pin for the head of the valve rod, and two slide blocks, the whole being generally forged in one piece. The slides are fitted with gunmetal faces above and below (Fig. 221). The pin itself is kept as short as possible, that the forked ends of the eccentric rods should not be too wide.

Allowable pressure on the link block pin = P = 550 to 1,000 lb. per square inch.

Allowable pressure on the faces of the link block = P = 220 to 350 lb. per square inch for merchant, 350 to 600 lb. per square inch for war vessels.

Each of the two quadrant link bars is of such strength that, when half the load of the valve rod $\frac{Q}{2}$ is thrown upon its centre, the bending strain is within moderate limits. (See Fig. 223).

The section of the sliding block is worked out from—

$$b \times h^2 = 0{\cdot}75 \frac{Q \times a}{S}$$

The following are the values of permissible stress :—

s = 3,500 to 8,500 lb. per square inch for merchant vessels.
s = 5,500 „ 10,000 „ „ war „

Also $h = 2{\cdot}5$ to $3b$, from which, with the above data, the values of h and b may be determined.

Eccentric Rod Pins.—These are generally forged in one with the link bar, and made without outer collars. The pressure on these pins should be from 650 to 1,050 lb. per square inch. Diameter of pin = about 1·2 to 1·3 × length of pin. The pin or pins for the reversing links are placed either in the centre of the quadrant, or may be simply an extension of the eccentric rod pins, but fitted with outside collars. The pressure upon them is calculated from the power transmitted through the reversing levers and links. (See Fig. 220, and page 251.)

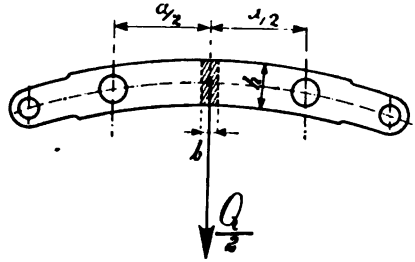


Fig. 223.

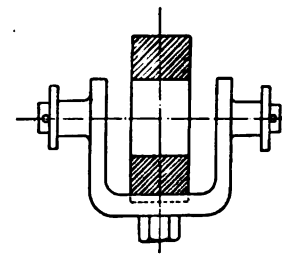


Fig. 224.

Solid Slotted Links.—These are still largely used for the engines of small screw steamers, and almost universally for engines of paddle steamers. The dimensions are determined in the same way as those of the double-bar quadrants. It will be apparent from its construction that the slot link is not suitable for transmitting heavy loads. The pins for attaching the reversing links may be fitted to the middle or to one end of the link in the same way as described above. (See Figs. 224 and 225.) The link block is generally formed of a single forging, the sliding surfaces of which are sometimes faced with white metal, or fitted with special strips.

§ 122. **Eccentric Rods.**—The bearings are generally similar in design to those for the connecting rod, and bushed with gunmetal, but without any white metal. The brasses may be square or round (Fig. 226). The fork of the "ahead" driving rod is usually made symmetrical, and that of the "astern" driving rod more or less one-sided. Sometimes the "astern" rods are joggled just above the

eccentric straps, thus making their forked ends also symmetrical. For calculating the dimensions of the forks see § 102, page 195. Whenever it is possible, the ahead and astern running eccentric rods are both made the same size, to avoid the necessity of carrying more than one spare rod of each sort. The rod is secured to the eccentric strap by a flange and two set screws or pins; in large engines the rod and the upper part of the strap are seldom forged in one piece, owing to their size and the difficulty of erecting them. Round rods are simplest, but flat rods are generally used in the engines of warships.

The allowable stress in the eccentric rod immediately below the fork is 1,500 to 3,500 lb. per square inch.

The cross section at the bottom of the rod = about 1·8 to 2 × cross section at the top (for long rods). Cross section at the bottom of the rod = about 1·4 to 1·6 × cross section at the top (for short rods).

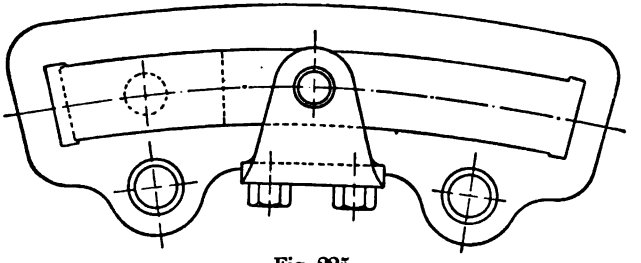


Fig. 225.

Length of the Eccentric Rods.—They should be as long as possible, as there is usually plenty of space for guiding the valve rod above the link.

The allowable stress in the fork bearing bolts, and in the set screws attaching the rod to the strap, is usually 2,000 to 5,500 lb. per square inch, according to the type of ship.

§ 123. **Eccentrics and Eccentric Straps.**—As a rule the eccentric or eccentric sheave is made as small in diameter as possible, and placed, if it can be arranged, on the shaft; but often it is impossible to avoid putting some or all the eccentrics on the shaft couplings. The ahead and astern eccentrics are either made of the same width, or the latter may in some cases be made slightly narrower. In other respects the same rule holds good as for the other parts of the valve gear, viz., that all the parts should as far as possible be of the same size, because they are cheaper to make, and fewer spare parts need be carried. In smaller engines the ahead and astern eccentrics are often cast in one piece. The eccentric sheaves are nearly always in two halves, and the

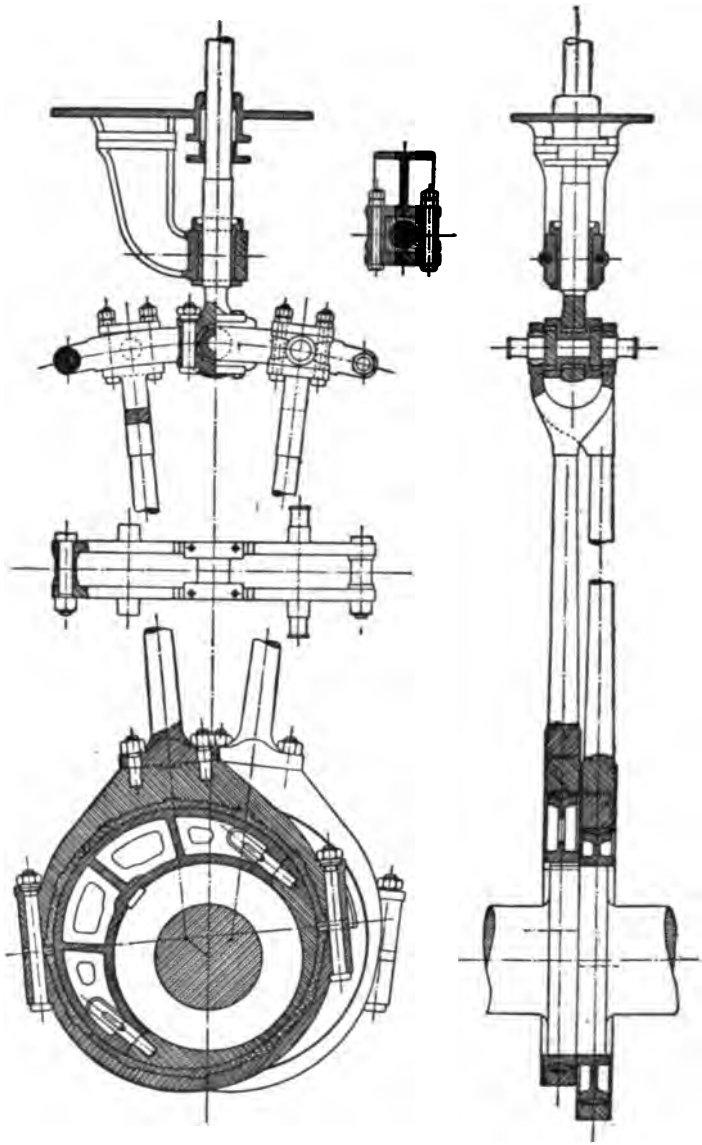


Fig. 226.

bolts bolting them together are made as strong as possible, and at least as large as the eccentric strap bolts. The bolts are secured by nuts, or, if room for these is wanting, by cotters. (See Figs. 226 and 227.)

Materials for Eccentric Sheaves.—In cargo boats both halves are made of cast iron, especially if the eccentrics are large, and the shaft small. This happens especially in engines where some of the eccentrics are fitted on the couplings and others on the shaft, but where all the straps are of the same size. In other cases the larger half is made of cast iron, the other half of cast or forged steel. Cast steel is now often used for both halves.

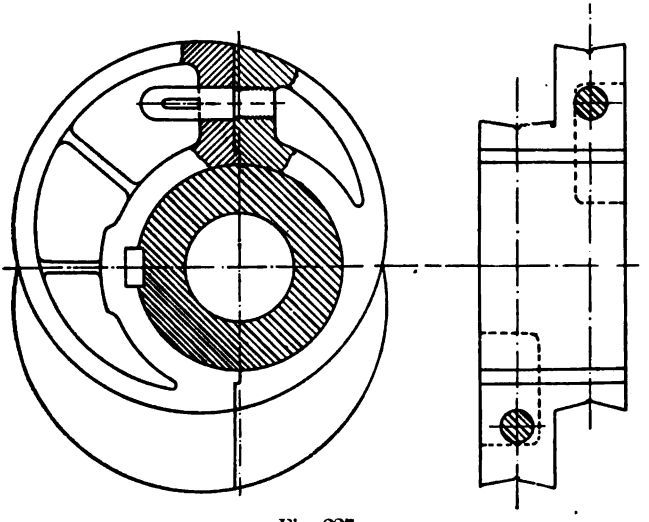


Fig. 227.

Size of Eccentric at Minimum Cross Section (see Fig. 230).—$s \times b = a \times$ cross section of both eccentric bolts.

For cast iron $a = 3{\cdot}5$ to 5.
,, cast steel $a = 1{\cdot}8$ to $2{\cdot}5$.
,, forged steel $a = 1{\cdot}5$ to $1{\cdot}8$.

As a rule the two halves of the eccentric have a groove and tongue on the jointing face, that they may be turned with accuracy. The eccentric must be most carefully fitted to the shaft, to preclude any possibility of its working loose. The key is made very wide, and is generally fitted in the crown or centre of the larger half of the eccentric. A set screw is added, to prevent the eccentric from moving sideways. Sometimes, with the same object, the ahead and astern gear eccentrics are bolted together, so that the key of the one eccentric acts as a stop to the other eccentric. This arrangement also distributes the strain over both keys.

The diameter of the eccentric having been determined from the cross section at the crown, the width is determined from the maximum pressure on the bearing surface, which may be taken at from 70 to 140 lb. per square inch, according to the weight and space available.

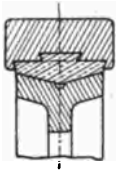


Fig. 228.

The greater the circumferential speed of the eccentric, the lower the value selected for the allowable pressure between the surfaces, other conditions being equal. The higher the speed of the working surfaces, the greater the friction per unit of time, and consequently the greater the chance of overheating. To prevent the eccentric strap slipping sideways, the face of the sheave is either made conical from either side inwards, or projecting edges are turned either on the eccentric or on the strap. The drawback to the latter arrangement is that the edges are apt to wear away, and the eccentric or strap then becomes useless (Fig. 229).

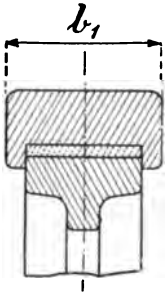


Fig. 229.

§ 124. **Eccentric Straps.**—In small vessels these are often made of gunmetal; in large vessels almost universally of cast steel with white metal linings, and occasionally of cast iron with gunmetal linings. The latter are heavy, and not durable. In very light warships the straps are often made entirely of wrought iron or steel, and sometimes the rod and top strap are forged in one piece, as in locomotives. For the connection between the top strap and the rod, see § 122, page 228.

The top half of the strap, where it is bolted to the rod, must be thick enough in the middle not to bend or pull away from the sheave. h, Fig. 230, should therefore be as deep as possible.

Moment of resistance per unit stress at h (Fig. 230)—

$$w = \frac{b_1 \times h_2}{6} = \frac{Q}{2} \times a_1 \times \frac{1}{s}$$

s being = 4,250 to 8,500 lb. per square inch for wrought iron or steel or cast steel, and = 1,140 to 4,250 lb. per square inch for cast iron. In some engines s is as much as 14,000 lb. per square inch for cast steel, but with such a stress the eccentrics are apt to run hot, because the side or fork of the strap tends to close in. If the upper half of the strap is strong enough to resist this action, the lower half need only act, more or less, as a connecting band. As a rule this half is made thicker at s_1, Fig. 230, near the crown, than at the sides.

Eccentric straps of cast steel, whose cross-sectional area in the lower

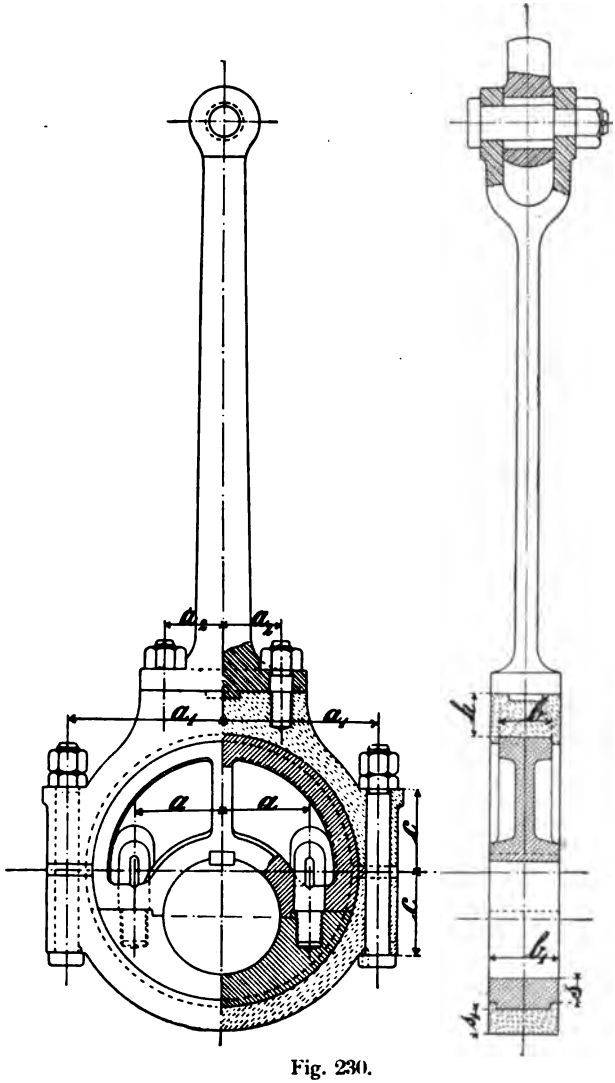


Fig. 230.

half is double that under the thread of the two strap bolts, will be found strong enough. Generally—

$$s_1 \times b_1 = 2 \text{ to } 2{\cdot}5\text{F for cast steel.}$$
$$s_1 \times b_1 = 1{\cdot}8 \text{ to } 2\text{F for wrought steel.}$$

F being the area under the threads of the two strap bolts.

The eccentric strap bolts are put as close together as possible, even

though it entails an increase in their length. The bolts should be large enough to withstand the load of the valves, Q, and the resulting stress must not exceed 2,100 to 5,500 lb. per square inch (according to the type of vessel). They should be fitted with lock nuts or ring nuts and pins, a fine thread being advisable. (See Table 72, page 609.)

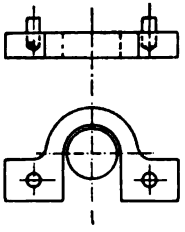


Fig. 231.

Gunmetal fitting strips are fitted in the joint between the top and the bottom halves of the eccentric strap. They are usually made so that they can be taken out from the sides, the metal inside the bolt holes being cut away for this purpose (Fig. 231). Steady pins should also be provided.

As with all bearings fitted with white metal, the white metal must be held in position at each side by a projecting edge, and also by dovetailed grooves or slots.

§ 125. **Concluding Remarks.**—Only Stephenson's link motion has been here considered; but the different parts of other systems of valve gear may be designed in the same way, the load upon the valves Q being taken as a basis.

Bed-plates.

§ 126. **The Bed-plate** consists of as many transverse girders as there are bearings, these transverse members being connected by longitudinal members. In smaller engines (up to about 1,000 I.H.P.), for cargo or

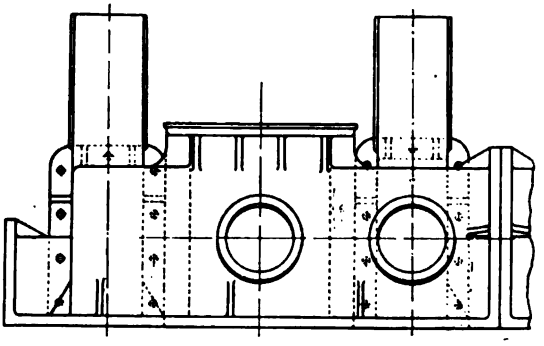


Fig. 232.

passenger steamers, tugs, pinnaces, &c., the whole bed-plate is cast in one piece, even when there are several cranks; the cross section of the transverse members is T shaped, ⌐ or ⌐⌐ shaped, and the

cross section of the longitudinal members ⌐└ or ┘┌└ shaped. Thickness of the metal of the bed-plate (δ), if of cast iron, is—

$$\delta = \frac{d}{30} + \cdot 5 \text{ inch } (d \text{ being the diameter of the shaft in inches}).$$

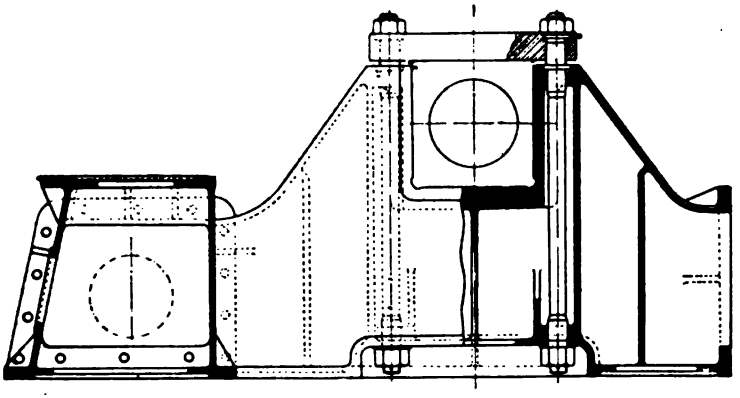


Fig. 233.

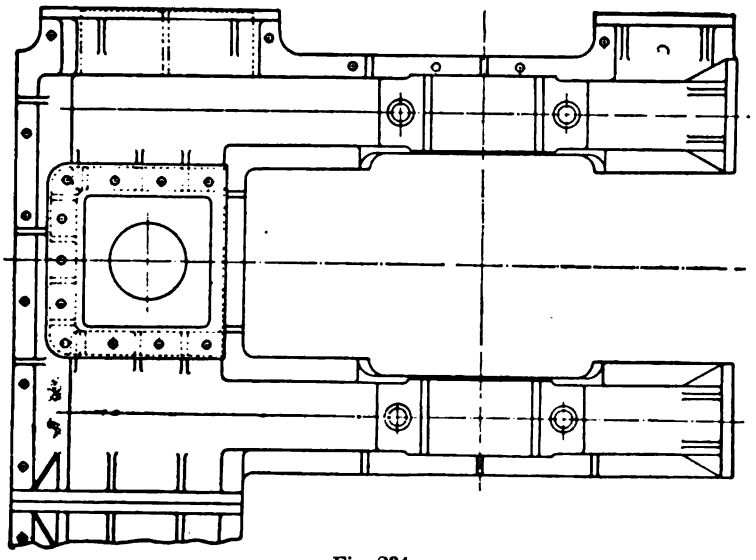


Fig. 234.

In engines for large merchant vessels a separate bed-plate is made for each crank. The bed-plates each contain two transverse members, and are bolted together by means of flanges. (See Figs. 232 to 234.)

The cross section of the transverse and longitudinal members is channel-shaped thus ⎽⎍⎿. The thickness of the metal, if of cast iron, is, as before—

$$\delta = \frac{d}{30} + \cdot 5 \text{ inch.}$$

The longitudinal members on one side are generally replaced by the condenser, which is bolted to the bed-plate. (See § 151, page 277.) The thickness of the flanges for cast iron, allowance being made for planing the casting afterwards, is—

$$\delta_1 = 1 \cdot 9 \text{ to } 2\delta.$$

The flanges are connected by strong fitted bolts.

In the engines of modern quick-running steamers the bearing frames are often made of cast steel. As it is not desirable to have very complicated castings in steel, these transverse members are made separately and joined by distance pieces. The cross section of both is made ⎽⎍⎿ shaped. In the case of steel castings, large openings are provided in the sides, to enable the core to be rapidly removed after casting, on account of the rapid shrinking of this material. (See Figs. 102, 105, 107.) For thickness of cast-steel bed-plates, see below.

In *modern warships* the framing to carry the bearings is always made of cast steel in separate castings, and these are joined either by distance pieces, or, preferably, as is generally done, by two side pieces running from end to end, having a section shaped thus ⎿ or thus ⊏ (Figs. 235 to 239). The engine of a small cruiser shown at Fig. 90, has a bed-plate of the first-named type. The photograph is taken in the erecting shop; the cast-iron supports which are placed below the cast-steel cross girders, only serve for the erection of the bed-plate, and are replaced in the ship by the actual engine seating. The longitudinal girders at the sides are of cast steel, or, in torpedo-boats or other light craft, of wrought or rolled steel. The cross section of the transverse members for the bearings is shaped thus ⎽⎍⎿, and made much deeper in the middle than on the outside; firstly, because of the extra strength thus secured; secondly, because, when cramped for headroom for the engine, the shaft must lie as low as possible in the ship; and lastly, because the plating forming the engine seating under

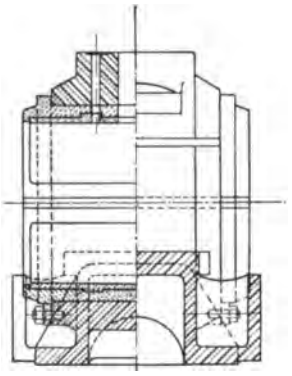


Fig. 235.

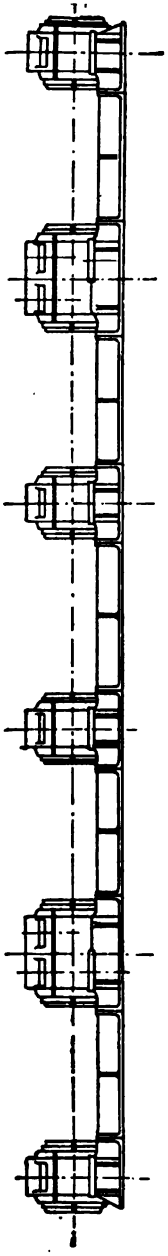


Fig. 236.

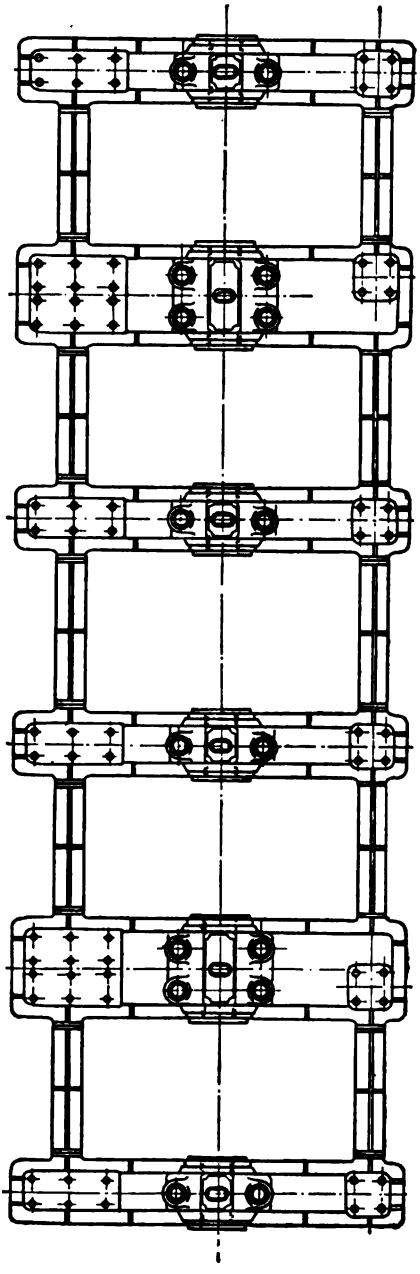


Fig. 237.

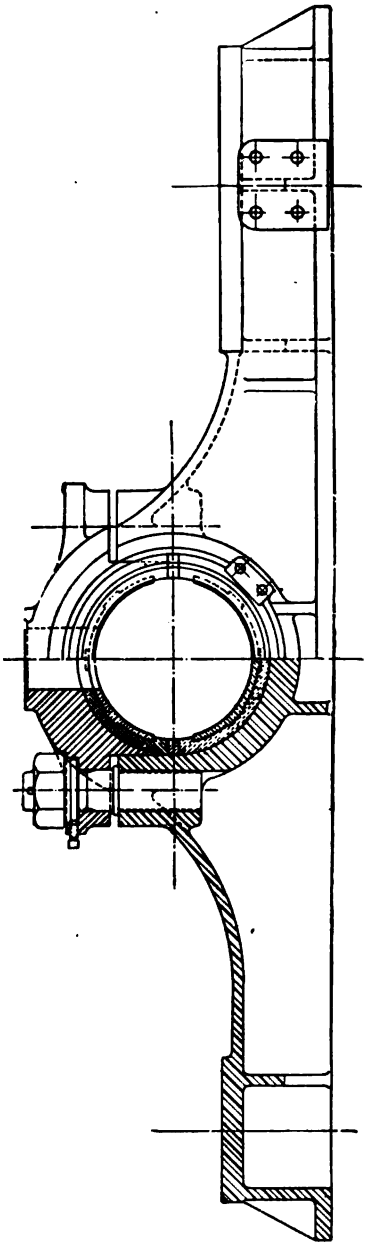


Fig. 238.

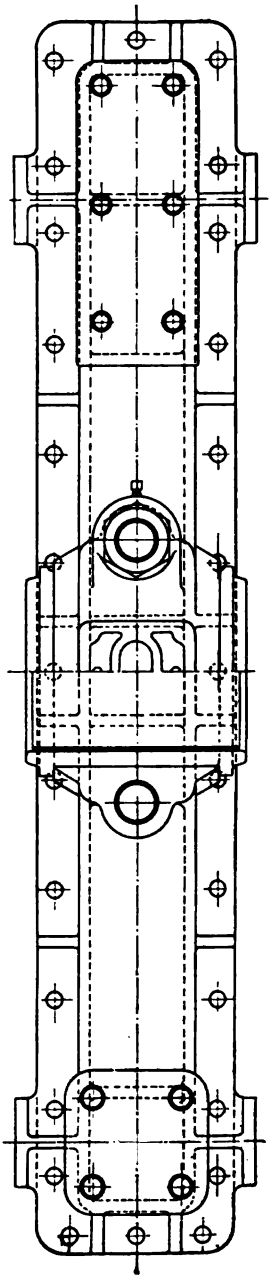


Fig. 239.

the side bearers should be as large, strong, and deep as possible (see § 359, page 616).

Thickness of the cast-steel bed-plates, $\delta = \frac{d}{50} + \cdot 6$ inch.

„ „ flanges, $\delta_1 = 1 \cdot 6\delta$.

In merchant ships the *height of the centre of the shaft above the under side of the bed-plate*, in steel and cast-iron bed-plates, is such that in its lowest position the cap of the connecting rod does not come below the bottom of the bed-plate. The latter either rests on fitting strips of hard wood, and the under side is then left rough, or it rests upon cast-iron or steel fitting strips, in which case the under side of the bed-plate is usually planed up.

The *foundation bolts* generally pass through the fitting strips, the spaces between the latter being filled in with hard wood.

Diameter of foundation bolts—

$\frac{3}{4}$	to	$\frac{7}{8}$	inch for shafts under 4 inches diameter.							
$\frac{7}{8}$	„	1	„	„	from	4	to	8	inches diameter.	
1	„	$1\frac{1}{4}$	„	„	„	8	„	16	„	„
$1\frac{1}{4}$	„	$1\frac{3}{4}$	„	„	„	16	„	24	„	„

In bed-plates with flanges running all round them the distance of the bolts from each other is about ten times their diameter.

§ 127. **Holding-down Bolts.**—The holding-down bolts must be sufficiently strong to withstand the upward vertical component of the

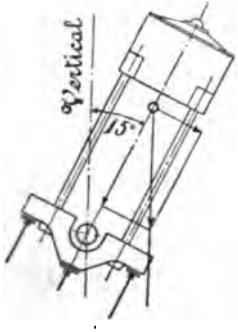


Fig. 240.

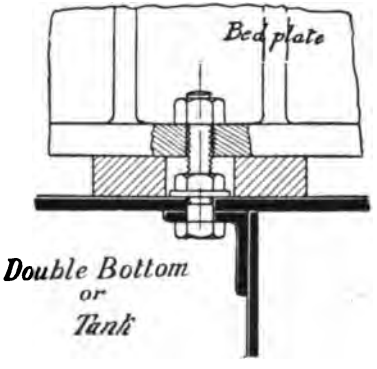


Fig. 241.

moving parts (see page 60), allowance, of course, being made for the weight of the engine. They must also hold the bed-plate in position firmly enough to withstand the horizontal component of the moving

parts, and the tendency to slide, due to the weight of the engine, when the ship rolls up to say 15° from the vertical. (See Fig. 240.) The holding-down bolts are usually roughly and carelessly taken out, and should therefore be as strong as possible. In engines whose centre of gravity lies very high up, it may so happen that when the ship rolls the vertical line dropped from this centre falls beyond the bed-plate, and the bolts must then be strong enough to prevent the engine tilting over on one edge of the bed-plate. In the engines of torpedo-boats, side projections are sometimes provided on the bed-plate to counteract this tendency. (See Fig. 251.) If the bed-plate is immediately above the water tanks in the double bottom, the joint between the bolts and the top plate must be made watertight. Either the head of the bolt projecting into the tank must be carefully packed, or a nut must be fitted in between the top plate and the flange of the bed-plate (Fig. 241).

Definite rules cannot be laid down for the thickness of the longitudinal and transverse members of the bed-plate, because this depends in the main upon the solidity of the engine foundation. The transverse members must always be of sufficient strength, so that, when the maximum load is thrown upon their centre, the bending strains set up in them are very small. As a rule their height is such that, as mentioned above, the cap of the connecting rod in its lowest position does not come below the bottom of the bed-plate.

§ 128. **The Longitudinal Bearers,** together with the longitudinal plates of the engine seating, must be strong enough to withstand the turning moment of the engine without appreciable distortion. (See § 35, page 82.) Sufficient strength to resist these forces can only be obtained by having a strong foundation, and a good connection between the upper ends of the columns. For this reason, in warships especially, the longitudinal members of the bed-plate are low and small, and serve principally, during the erection of the engine, as distance pieces for staying the transverse members, the engine seating itself affording the requisite longitudinal rigidity.

§ 129. **Main Bearings.**—For the diameter of these, see § 109, page 208. The length is determined according to the allowable pressure upon the bearing surfaces, due to the maximum load.

The allowable pressure on the main bearings should not exceed the following:—

Ordinary freight steamers	200 to 225	lb. per sq. in.
Passenger boats and quick-running steamers	225 „ 300	„ „
Ironclads, large cruisers	250 „ 350	„ „
Small, light cruisers	350 „ 400	„ „
Torpedo-boats, steam tugs, &c.	400 „ 550	„ „

In engines with several cranks it is assumed that the maximum load of the H.P. cylinder acts upon each crank, and the total length of the main bearings is such, that the maximum load, divided by the total pressure on their surfaces, does not exceed the allowable pressure given in the above table. The main bearings are generally all made of equal length. The ends of the brasses approach the cheeks of the crank as near as possible, but sufficient play must be allowed to prevent the

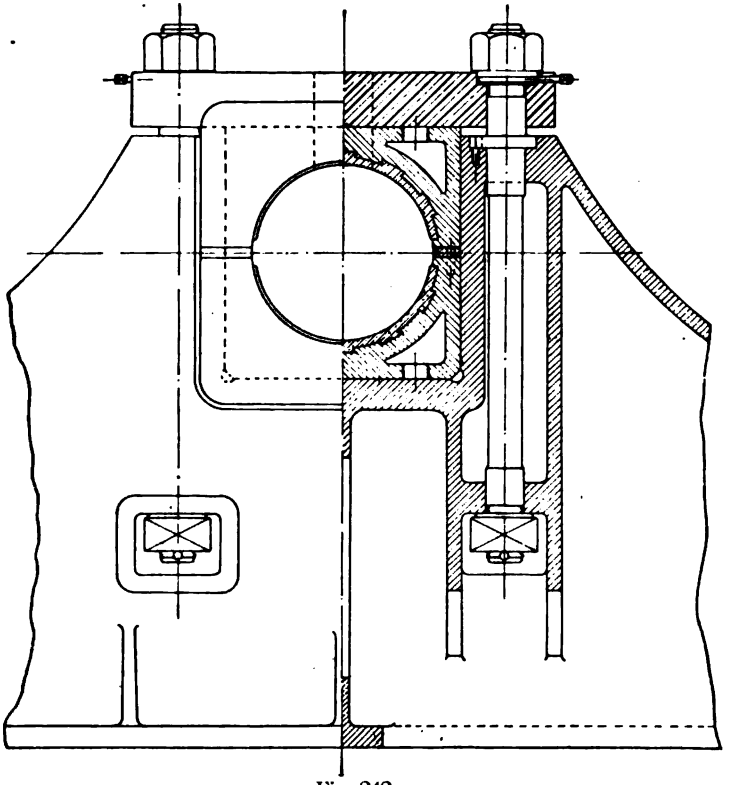


Fig. 242.

cheeks fouling the brasses as the thrust block wears or the crank shaft expands with heat.

Brasses.—As a rule the lower brass is made round at the base, and the upper one square on the top. In large cargo and fast passenger steamers the lower brasses are frequently made with square bottoms, while occasionally in warships both upper and lower brasses are made round.

Square Top and Bottom Brasses (Figs. 242 and 243).—Both brasses should be fitted perfectly tight into the bearing block. If this is done

there is less fear of their wearing loose than if rounded brasses are used. If the bearings are large the brasses are made hollow, and water is circulated through them.

The material of which they are made is usually cast iron or gun-metal, lined with white metal held in place by dovetailed grooves and recesses. The brasses are tinned before the white metal is run in. Sometimes the white metal is only fitted into the brasses in longitudinal

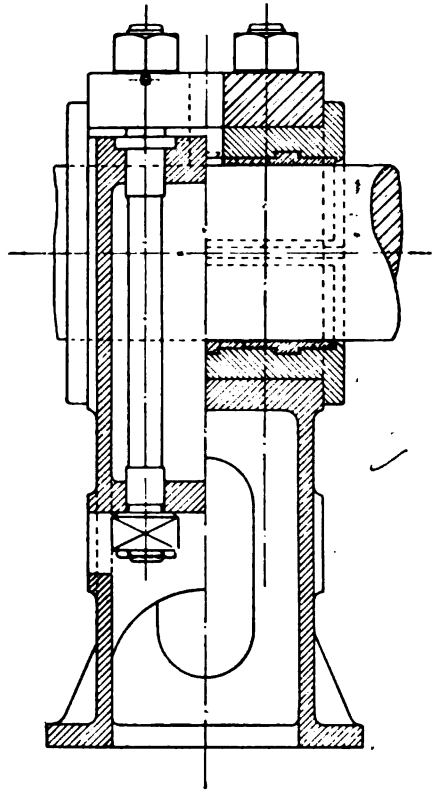


Fig. 243.

strips. Between the two brasses intermediate packing pieces or liners of gunmetal are placed, and held in position by set pins, and under these again are laid a few thin pieces of metal plate, which are removed as the wear on the bearing is taken up. To get all the bearings exactly in line, they are machined up in place after the bed-plate has been bolted together. The white metal in the bearings is left in the rough, and both ends of each bearing are fitted with a wooden disc,

in the centre of which is a piece of thin metal plate with a very small hole in it. The metal centres are shifted until a light, placed opposite the hole in the centre of the shaft at one end of the engine, can be sighted through all the little holes simultaneously. The hole in the thin metal plate then forms a centre, from which a circle can be struck on each brass, for boring out the white metal.

Round brasses have the advantage that they can be taken out without removing the shaft. For instance, if a bearing is working hot, the lower brass can be turned round and drawn out, the surface scraped, and the oil channels made deeper. To effect this, the shaft is slightly lifted, and the brass drawn out with a bent hook (Fig. 245). If both the upper and lower brasses are round, the packing pieces or liners between them must be held tight by lugs on the cap, to prevent the brasses turning round.

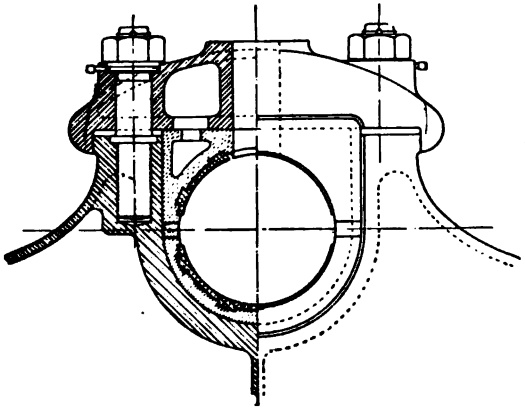


Fig. 244.

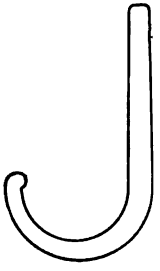


Fig. 245.

§ 130. **Main Bearing Caps.**—If the upper brass is round, the cap is made of cast iron or cast steel; if square, a thick wrought-iron or steel plate or slab is generally used. For dimensions see page 245. A hole about $4\frac{1}{2} \times 3$ inches, large enough to admit the hand, is made in the cap and the upper brass, through which grease can be applied to the bearing, and its temperature felt. Over this hole an oil box is usually fitted with two divisions, one having an opening direct to the hole, and the other having a worsted syphon-feed arrangement. The cap and upper brass are often joined by special screws, so that both can be removed together. Tapped holes are also provided in the cap and in both the brasses for eyebolts.

§ 131. **Main Bearing Bolts.**—Of these there are two for small, and four for large bearings. Where it is possible, bolts are used with

nuts at either end, and the lower one is often a square-headed nut, which cannot be turned (Figs. 242, 243). Less often the bolts have heads at their lower ends; in any case they should be as easily removable as possible. Bolts with nuts above and below are held tight in the block by a collar countersunk in the upper surface under the block. Care should be taken that the latter is well stiffened where the bolts go through it. (See Figs. 242, 243.)

With smaller bearings, studs up to about 3 inches diameter are used, but they have the disadvantage that they are difficult to replace, especially if they break off below the collar. Both bolts and studs are made with fine threads (see Table No. 71, page 608), and in larger engines ring lock nuts are used for the cap, and double nuts in smaller engines. Ring nuts

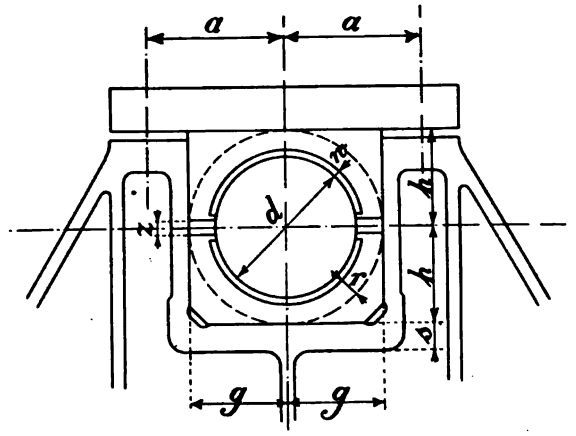


Fig. 246.

should be graduated so that they can be screwed up and carefully set. Tapped holes should be provided in the top of the bolts for eyebolts.

§ 132. **Dimensions of Main Bearings** (see Fig. 246).—Distance of the bolts from centre of the bearing—

$a = 0{\cdot}85$ to $0{\cdot}9d$, if the brass and bed-plate are of cast iron.

$a = 0{\cdot}75$ to $0{\cdot}85d$, if the brasses are of gunmetal and the bed-plate is of cast steel.

Width of the brass - - - $g = \frac{d}{2} + r + w$.

Depth of the brass - - - $h = 1{\cdot}05g$ to g.

Thickness of the white metal $w = {\cdot}20 + \frac{d}{35}$ inches.

Thickness of the distance pieces $z = {\cdot}20 + \frac{d}{15}$ inches.

Thickness of *round brasses*, omitting the thickness of the white metal—

$r = 0{\cdot}07$ to $0{\cdot}09d + {\cdot}125$ inch for bronze.
$r = 0{\cdot}11$ „ $0{\cdot}12d + {\cdot}20$ „ cast iron.

Thickness of bed-plate below the brasses—

$s = 0{\cdot}20$ to $0{\cdot}28d$ for cast iron.
$s = 0{\cdot}12$ „ $0{\cdot}16d$ „ cast steel.

Strength and Size of Main Bearing Bolts, &c.—If the maximum load on one crank be divided between the bolts in the two adjacent bearings, the stress in each should not exceed—

s = 3,000 to 4,250 lb. per square inch in merchant vessels.
s = 4,250 „ 6,500 „ „ warships.
s = 6,500 „ 7,750 „ „ torpedo-boats.

If there is one bearing between two cranks, it is generally fitted with four bolts; if there are only two bolts, they are made proportionally stronger.

§ 133. **Thickness of Caps.**—These are generally taken as beams, supported at points represented by the centres of the bolts, with a uniformly distributed load, the total of which is equal to half the maximum load $= \frac{P}{2}$. The bending stress is generally somewhat less than that allowed for the connecting-rod cap. If there is only one bearing between two cranks, the cap is made from $\frac{3}{2}$ to $\frac{4}{3}$ times as strong as in the other bearings. Width of the cap b = about $0{\cdot}6$ to $0{\cdot}9 \times$ length of the bearing.

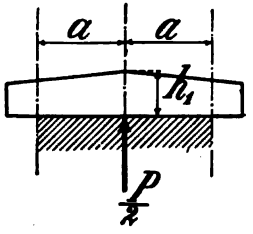


Fig. 247.

The depth of the cap is obtained from the above values of s by the formula—

$$s = \frac{\text{Bending moment}}{\text{Modulus of section}} = \frac{\frac{P}{2} \times 2a \times \frac{1}{8}}{b \times \frac{h_1^2}{6}}$$

$$\text{whence } h_1 = \tfrac{1}{2}\sqrt{\frac{3P.a}{b.s}}$$

Engine Columns.

§ 134. **The Arrangement of the Columns** of an engine considerably affects its construction, and is characteristic of the type of the engine.

§ 135. 1. **Engines for Ordinary Small Merchant Vessels.**—The columns are of cast iron, either open and ribbed, or hollow and in one piece with the guides. Two columns are provided for each cylinder, or turned columns in front, and cast columns behind. The columns on one side are frequently cast in one with the condenser, and also sometimes with the bearings for the reversing gear and for the air-pump levers. With jet condensers, even the air pump is sometimes cast with the columns. The typical form is shown in Figs. 82 to 84.

Thickness of columns, whether cast hollow or ribbed $\delta = \frac{d}{30} + \cdot 5$ in.

„ flanges - - - - - - $\delta_1 = 2$ to $2 \cdot 2\delta$.

„ guide plate - - - - - $\delta_2 = 1 \cdot 5$ to $1 \cdot 7\delta$.

d being the diameter of engine shaft.

§ 135A. 2. **Heavy-built Engines for Large Merchant Vessels.**—Hollow cast-iron columns of square section are almost universally used, two to each cylinder. The back columns are almost always placed on the condenser, provided they fall within its length. For typical shape of column see Fig. 101. The guides are bolted on, and are generally water-cooled. (See § 107, page 205.)

Thickness of columns $\delta = \frac{d}{30} + \cdot 5$ inch.

„ flanges $\delta_1 = 2$ to $2 \cdot 2\delta$.

The flanges are strengthened with strong ribs or webs.

In large engines the tall columns are often forked at the bottom in shape like an inverted Y. This method of construction increases the stability of the engine, and the columns can be placed nearer to the shaft, because the crank can work freely between the two legs of the column. For typical shape, see Figs. 83 to 85, and 248, 249.

For facility in machining and in erecting the engine it is not desirable to cast the bearings for the reversing-gear or air-pump link gear, &c., in one piece with the columns. It is better practice to leave the requisite facings for these as well as for the draincock gear, reversing gear, &c. The columns are frequently fitted at the top with brackets

or flanges, to which the distance pieces are bolted for bracing the columns together. The hollow casting of the column is often utilised as an oil tank, and not infrequently, in the case of the L.P column, the exhaust steam is led through it direct to the condenser.

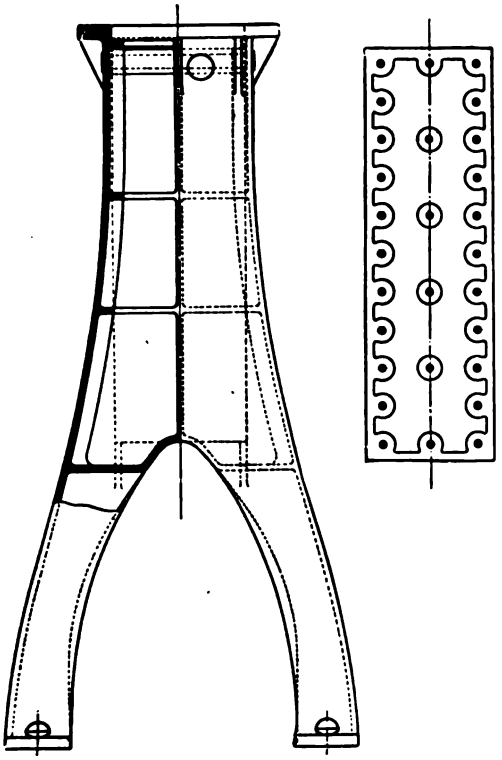


Fig. 248.

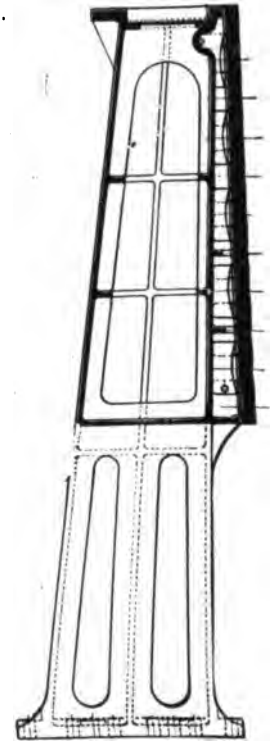
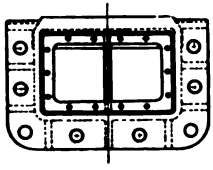
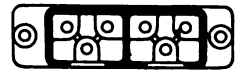


Fig. 249.





§ 136. 3. **Engines for Modern Fast Steamers and Large Warships.**—In these the condenser is generally separate, and is either of cast gunmetal or of sheet copper or brass, and is generally placed in the wing of the vessel. There are usually from two to four

long hollow or ribbed cast-steel columns to each cylinder. This is the type of construction adopted in H.M.S. "Powerful" and the new North-German Lloyd and Hamburg-American liners. (See Figs. 96, 97 to 99.)

Thickness of cast-steel columns $\delta = \frac{d}{40} + \cdot 5$ inch to $\frac{d}{35} + \cdot 5$ inch.

„ flanges for cast steel $\delta_1 = 2 \cdot 2$ to $2 \cdot 8\delta$.

d being the diameter of crank shaft in inches.

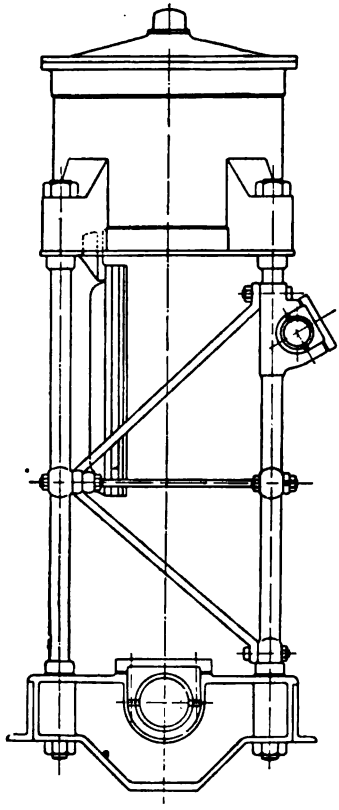


Fig. 250.

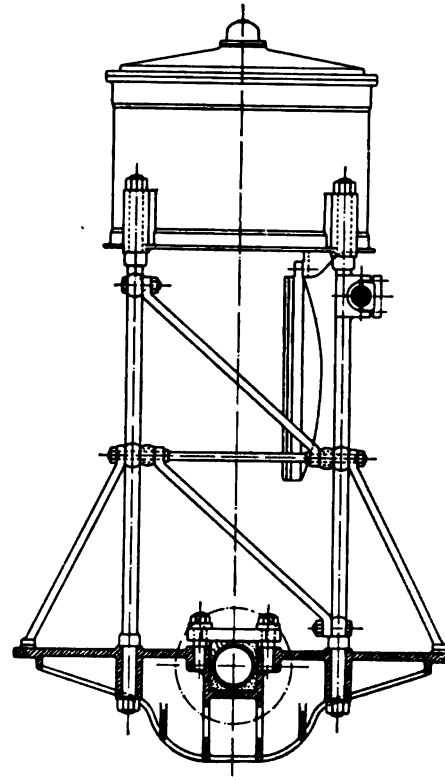


Fig. 251.

§ 137. 4. **Engines for Warships in general** (except Torpedo-boats).—These usually have on the front of the engine two wrought-iron, or steel columns for each cylinder, and on the side of the ahead guide, either one or two cast-steel columns, to which the cast-iron guide is attached. The reversing-gear bearings are either bolted

to the back columns or to the cylinder, or are fitted into forged projections on the front columns. The columns are fitted with top and bottom flanges. The thickness of the columns is the same as that given in § 136.

§ 138. 5. **Very Light Engines.**—In torpedo-boats, yachts, and steam pinnaces, the cylinders are usually placed on wrought-steel columns. These must be properly braced together in such a way as to absorb the strains coming on the engine framing. In larger engines of this type there are generally four columns to each cylinder, the two front columns being firmly connected to the corresponding back columns by diagonal ties. The columns are also braced diagonally in the longitudinal direction. For types of this arrangement see Figs. 72 to 75, and 250, 251. For drawings of engines supported on columns see Fig. 89, also "Arrangement of Main Engines," p. 106. For engines supported on inclined columns see Fig. 113, also Figs. 254, 255.

The columns generally have collars and screwed ends top and bottom. These are fixed with nuts into strong lugs on the cylinder and bed-plate. Sometimes flanges forged on the columns are used, instead of screwed ends and nuts. Even in large engines for merchant vessels the cylinders are sometimes supported on wrought-steel columns. This arrangement has the advantage of rendering the engine more accessible and easier of inspection.

§ 139. **Stresses in the Columns and Framing.**—These are produced by the following:—

1. Weight of cylinders.
2. Maximum thrust of connecting rod.
3. Pressure on the guides.
4. Pressure exerted by the reversing shaft.
5. Strains due to the expansion stuffing boxes on the receiver pipes and main steam pipe.
6. Strains set up by the rolling of the ship.

It is usual, for the sake of simplicity, to calculate the size of the columns only to withstand the tensile stress due to the maximum load on the piston. The stresses allowed are correspondingly smaller, and may be as follows:—

350 to	640 lb.	per sq. in.	for	cast-iron columns	for merchant vessels.
1,280 „	1,420	„ „		cast-steel „	fast steamers.
1,420 „	1,850	„ „		cast-steel „	warships.
1,700 „	2,125	„ „		wrought-steel „	„
„	7,000	„ „		„ „	torpedo-boats.

The latter stress is taken on the area at the bottom of the thread. In engines with cylinders supported entirely by columns the stresses can be more accurately calculated.

Example.—Calculation of the stresses set up in the columns and

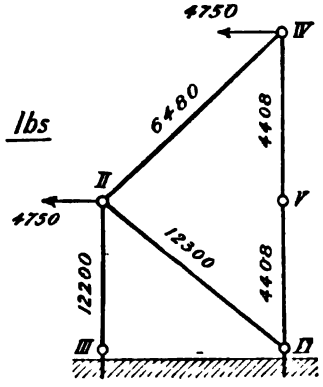


Fig. 252.

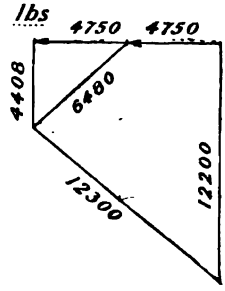


Fig. 252A.

diagonal bracing of a destroyer due to the pressure on the guides. For diagrammatic arrangement see Fig. 250. Let the maximum load on the piston be 76,000 lb., then the greatest horizontal force on the guide will be (if length of connecting rod = 4 × radius of the crank) $Q = \frac{76{,}000}{4} = 19{,}000$ lb. This occurs when the crosshead is at about the middle of its stroke. The forces on the upper and lower guide supports will therefore be $= \frac{Q}{2}$, and the force $\frac{Q}{4} = 4{,}750$ lb. will be transmitted to each side column at the upper and lower end of the guide (points I and II in Fig. 253). The horizontal force at the upper end of the guide cannot be taken by the member I II, as the latter would bend. On the other hand, the cylinder transfers the thrust to the point IV. The tie II V, however, cannot bear any great strain, or the column IV V VI would bend. Thus, to take up the horizontal thrust Q, there remain only the links shown in Fig. 252. The stress in the different members is represented by the polygon of forces, Fig. 252A. Assuming the direction of

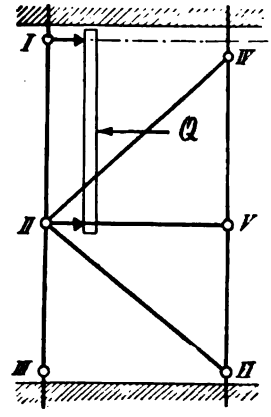


Fig. 253.

the forces to be as there shown, the stresses produced by the thrust on the guide will be as follows:—

Member	II IV, compression -	6,480 lb.
„	IV V and V VI, tension	4,408 „
„	II III, compression -	12,200 „
„	II VI, tension - -	12,300 „

If the engine is reversed, the direction of the thrust on the guide, and therefore of all the forces, is reversed.

§ 140. **Fixing the Columns.**—The columns are secured above and below by fitted bolts. If studs are used (and they cannot always be avoided in warships) a few accurately turned set pins must be used, to keep the columns in their places. The allowable tensile strain on the bolts at the bottom of the thread is from 3,500 to 5,600 lb. per square inch, according to the type of engine. It is best to make the bolts at the bottom end much stronger than those at the top, because they have to absorb the strains set up at right angles to the direction of the piston rod (thrust on guides, &c.). The steel columns, and also the horizontal distance pieces between the A-frames, are as a rule made somewhat short, and a fitting strip about $\frac{3}{16}$ to $\frac{5}{8}$ inch is used, which is accurately fitted while the engine is being erected.

Reversing and Turning Gear.

§ 141. **Reversing Shaft and Lever.**—The reversing shaft is parallel to the crank shaft, and carries a lever for the valve motion of each cylinder. The movement of this lever is transmitted by a link or links to the quadrant. The main lever for actuating the shaft is also fitted on to the reversing shaft, and is worked direct by the reversing gear, or by the reversing engine. This lever should be placed at about the centre of the shaft, so as to reduce the twisting stress upon the latter to a minimum. The bearings carrying the reversing shaft should be as close as possible to the various levers. This cannot always be managed, and the shaft is then exposed to bending as well as twisting stresses. Both must be allowed for when determining its diameter. For calculating the latter, the power required to reverse one valve motion, when the valve is exposed to the maximum steam pressure, should be taken as a basis.

Let r be the amount of eccentricity or $\left(\frac{\text{stroke of eccentric}}{2}\right)$ in inches.

R „ length of reversing lever in inches.

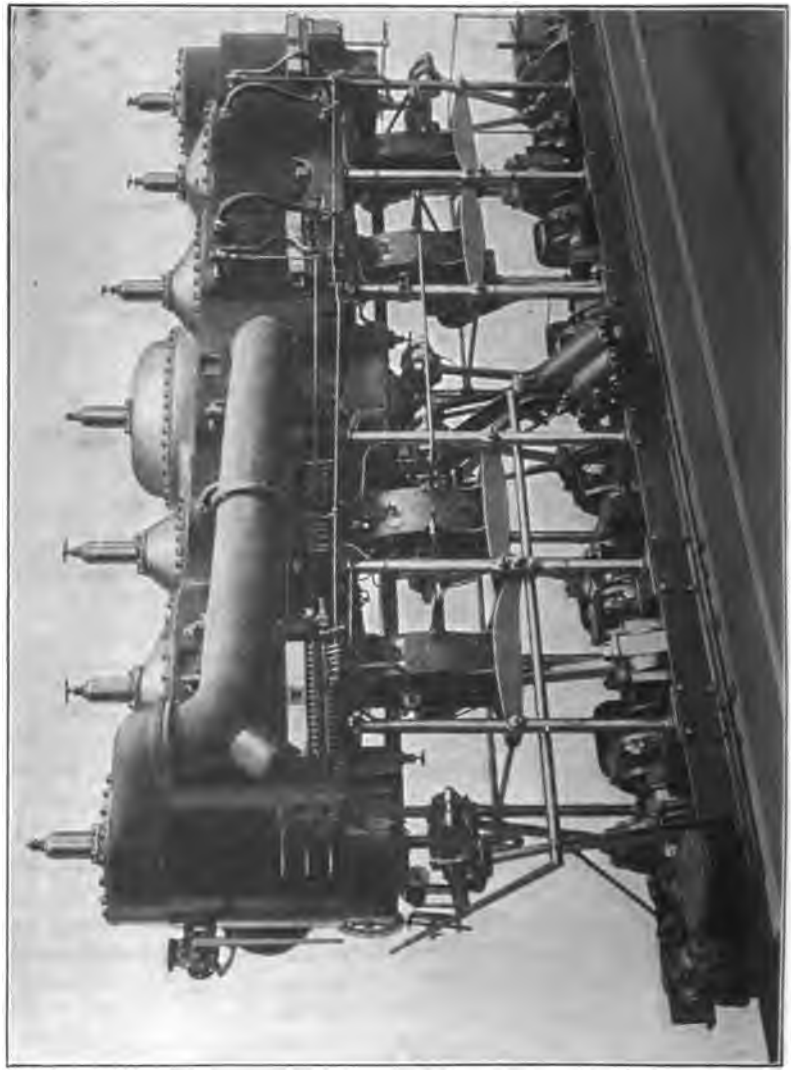


Fig. 254.

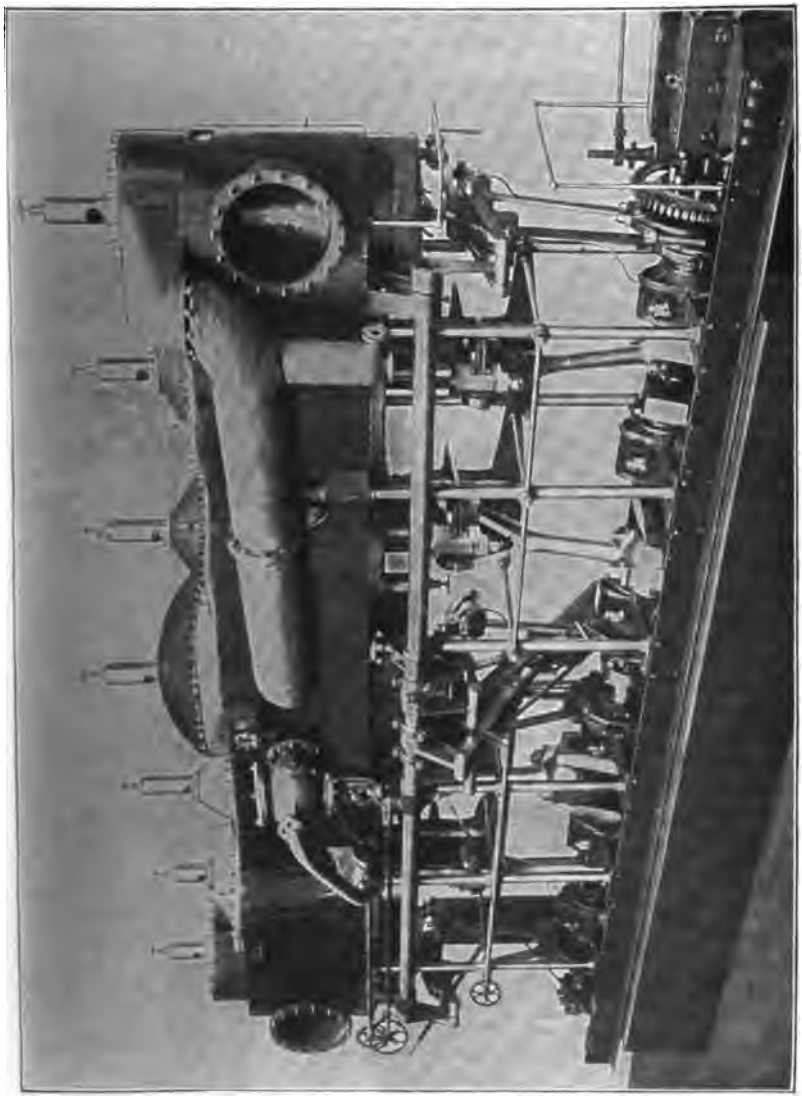


Fig. 255.

Let a be the distance between the eccentric-rod pins on the quadrant in inches.

Q „ maximum load on the valve rod in pounds (see page 222).

P „ maximum load required to reverse one valve motion in pounds.

Then $P = 2{\cdot}1Q\frac{r}{a}$. This power is divided between the two links connecting the lever to the quadrants.

The stress in the weakest part of the links is generally small, viz., about 1,200 to 1,500 lb. per square inch, and the stress in the reversing shaft-bearing bolts about 3,500 to 5,600 lb. per square inch. The pins

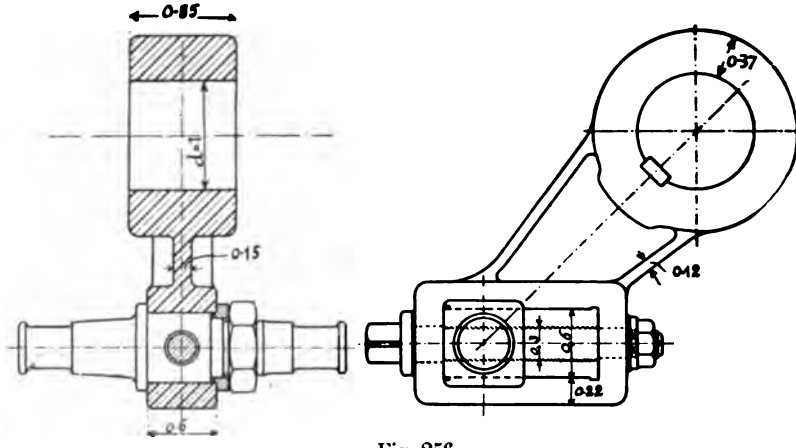


Fig. 256.

of the links, the length of which is greater than their diameter, should only be allowed to take a pressure of from 425 to 850 lb. per square inch. The diameter of the reversing shaft is therefore—

$$d = 1{\cdot}72\sqrt[3]{\frac{c \cdot P \cdot r}{k}}$$

Here $c = 1$ for engines with one crank.

$c = 1{\cdot}3$ „ two cranks at an angle of less than 90°.

$c = 1{\cdot}85$ „ three cranks „ „ 120°.

$c = 2{\cdot}4$ „ four cranks „ „ 90°.

The amount of twisting stress allowable k is generally about 3,500 to 4,250 lb. per square inch. If the shaft is subjected to a bending as well as a twisting stress, the above stresses should be somewhat reduced.

If weight has to be economised, the reversing shaft is usually made hollow.

In order to keep down the stress upon this shaft, it is desirable to make the levers working the links as short as possible. This may be done, for any given length of link, by increasing the angular travel of the reversing shaft; the latter, however, should not generally exceed 90°. (In engines of over 100 I.H.P. each reversing lever generally has a slot in which a slide-block works, and to which the reversing links are attached. The position of the block can be regulated by means of a screw.) (See Fig. 256.) By this arrangement the most economical cut-off in each cylinder can be easily obtained, and the power developed varied within certain limits (say about 12 to 16 °/₀). The slot is so designed that it is parallel to the valve rod when the engine is in full backward gear, so that the position of the valve gear is unaffected by the position of the adjusting block in the slots. (See Fig. 258.) If the angular travel of the reversing shaft or lever is 90°, the slot is at right angles to the valve rod, when the valve gear is in full forward gear. Hence the travel of the block in the slot is equal to the travel of the quadrants. The chord of the angle through which the lever travels,

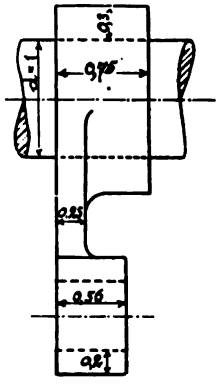


Fig. 257.

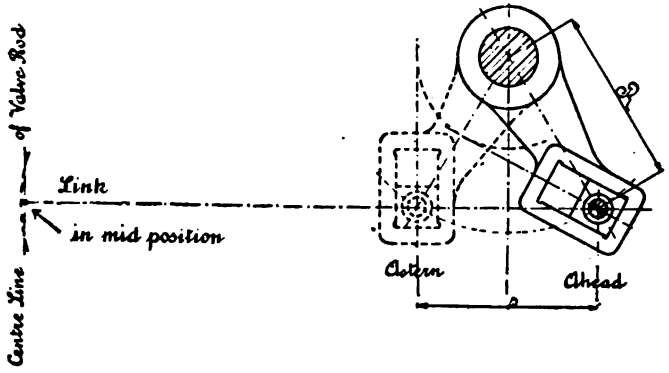


Fig. 258.

s, is generally equal to or somewhat less than the distance between the eccentric-rod pins on the quadrant. The reversing shaft and adjusting block are made of wrought iron or steel, the levers of cast iron, wrought iron, but more generally of cast steel.

If d is equal to 1, the usual dimensions for cast steel are given in Fig. 256. If cast iron be used, the dimensions of the levers should be increased by 10 to 15 °/₀, but the main reversing lever should always be considerably stouter than the auxiliary levers. If the levers are of forged steel (as is nearly always the case in engines for warships), they are generally of the shape shown in Fig. 257, and are either forced on to the reversing shaft under pressure, or shrunk on, and are secured by strong keys in both cases. To prevent any movement of the reversing shaft while the engine is running, some kind of locking gear is usually provided. If the reversing gear is worked by hand, a screw clamp may be used. If steam gear is used, such an arrangement is not possible, because if the ship has suddenly to be brought about, the clamping device takes too long to release. It is better to have a stop, against which one of the reversing levers, or a lever specially provided for that purpose, bears when the engine is in forward gear.

§ 142. **Method of Handling the Reversing Gear.**—Engines up to 150 H.P. may be reversed by means of a hand lever keyed direct to the reversing shaft. In larger engines, up to 500 H.P., the reversing shaft is generally worked by a hand wheel with either worm gear or screwed spindle; engines above 500 H.P. are generally provided with steam reversing gear, and a hand gear in addition, as a stand-by in case of need, the two being connected up to one another. Reversing engines are either "direct-acting" or what is known as "all-round" reversing engines.

§ 143. **Direct-acting Reversing Engines** (Brown's reversing gear).—These have a steam cylinder, and a brake or dash-pot cylinder filled with oil or water, in line with one another (Figs. 259-262). The brake or dash-pot cylinder is necessary to prevent a jerky or too rapid motion of the piston in the reversing engine, and therefore of the valve gear. In the reversing engine shown in the figures, the crosshead is connected to two levers on the reversing shaft by means of two links or connecting rods, the engine itself remaining stationary. A simple arrangement is to make the upper end of the piston rod project through the top of the brake cylinder, and attach the pin of the main reversing lever to it by means of an eye. The reversing engine does not remain stationary in this case, but can swing, at its bottom end, round a pin solidly connected to the bed-plate of the engine. The steam cylinder is worked by an ordinary D slide valve (with very small outside and inside lap), and the brake cylinder by a piston valve. (See Figs. 261 and 262.) The two sets of valves are interconnected and worked by a hand lever from the engine platform. As soon as both

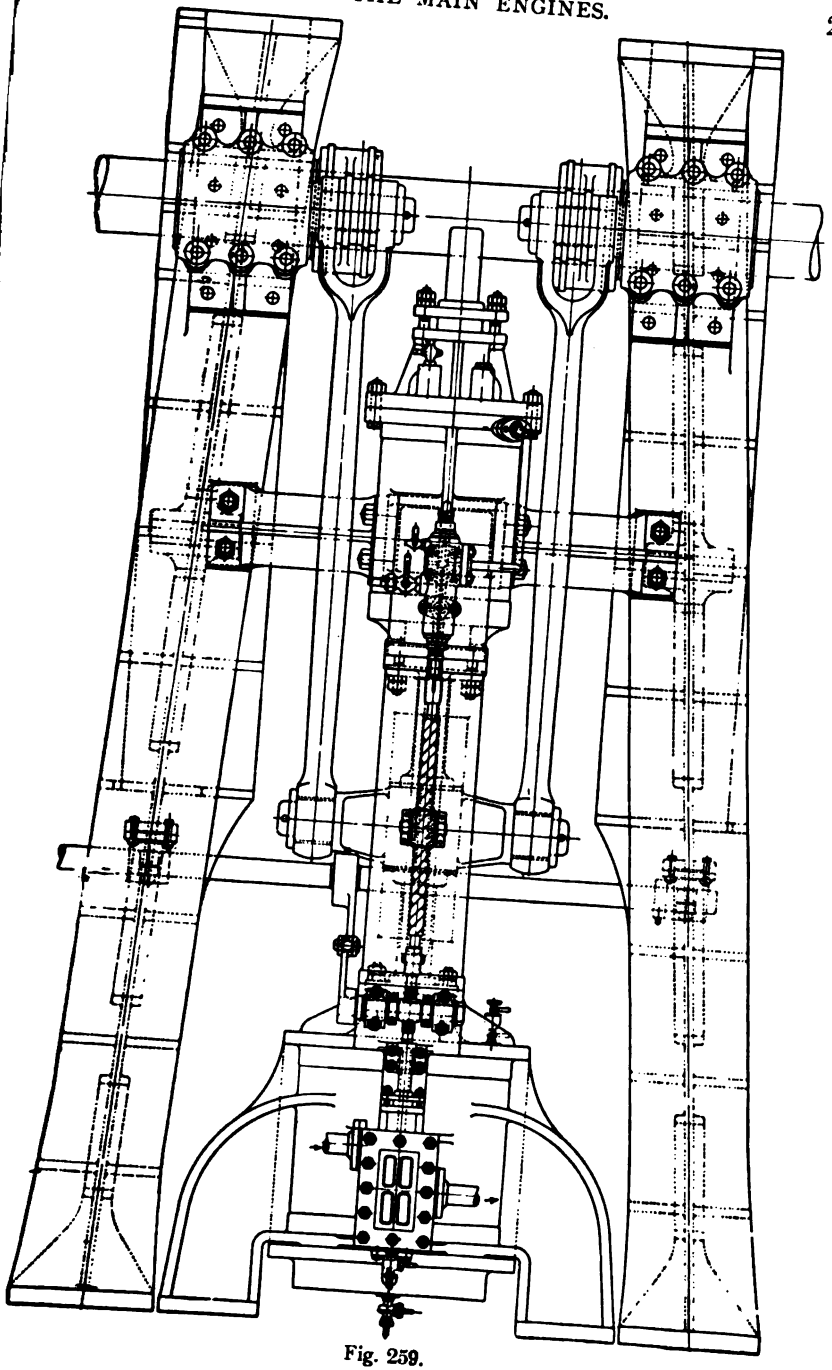


Fig. 259.

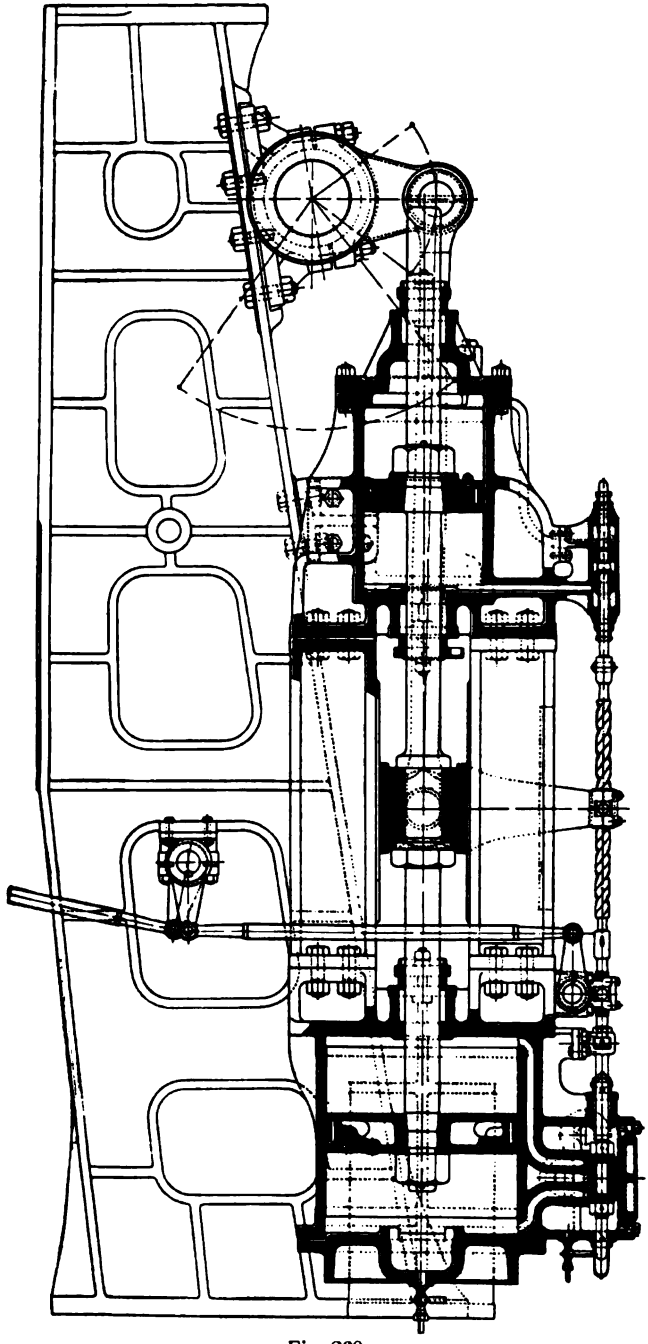


Fig. 260.

valves have moved, and steam enters the cylinder, the piston rod of the reversing engine begins to work, and carries with it the crosshead, to

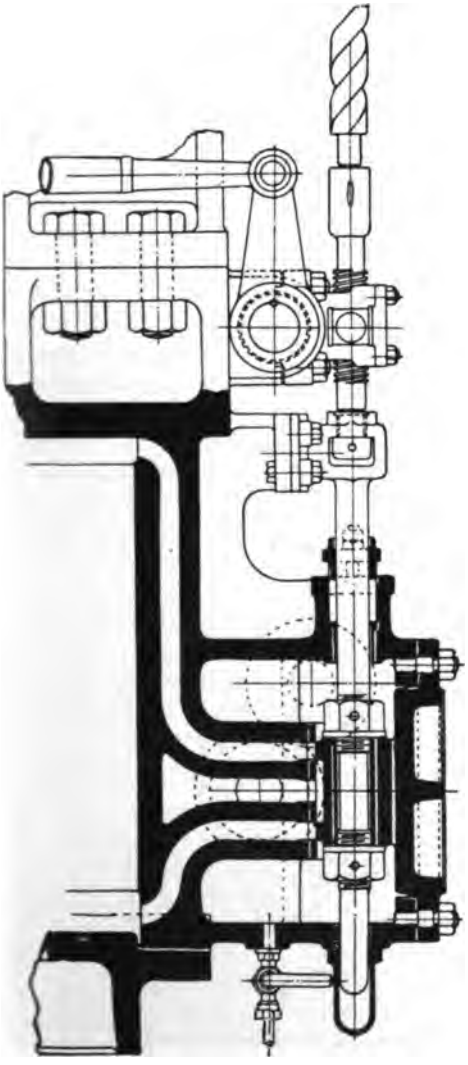


Fig. 261.

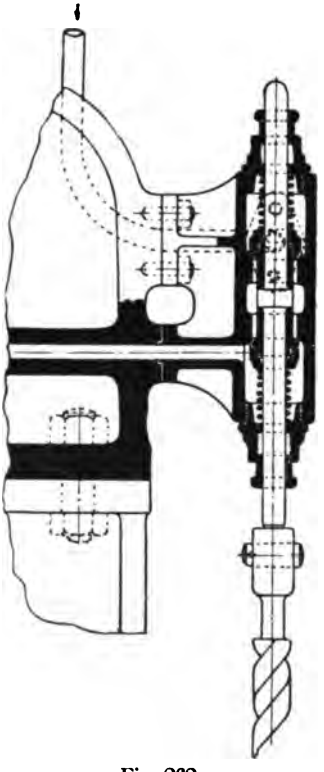


Fig. 262.

which the connecting links of the reversing gear are attached. To this crosshead is also connected an arm carrying a nut which is prevented from turning. It travels over a vertical spindle having a thread with

a very coarse pitch, and it is therefore able to rotate the spindle. The lower part of this spindle has also a fine thread, which works in a nut attached to the lever working the valve rod. The engineer having moved the reversing lever, the valve rod moves because the fine-threaded nut directly transmits the motion. The valves now being open, the piston and crosshead begin to move. The lever being now stationary, the valve spindle, caused to rotate by the coarse-threaded nut, travels as it turns in the fine-threaded nut which is fixed to the lever. The valve attached to the spindle travels with it, and thus comes into its mid position. This object may also be effected by means of a series of levers, in place of the screwed spindle. This is illustrated diagrammatically in Fig. 263. The slide valve of the steam cylinder is moved in the first instance by the hand lever; this sets the piston rod in motion, and as the point A is stationary, the valve has to return to its mid position.

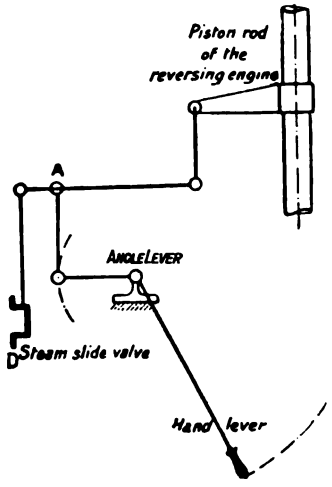


Fig. 263.

The brake cylinder may be used without separate valve gear, in which case a bye-pass, closed partly or wholly by a valve or cock, connects the two ends of the cylinder. Occasionally the brake piston has a small hole in it, through which the water can pass from one side of the piston to the other. The piston of the reversing engine is generally designed

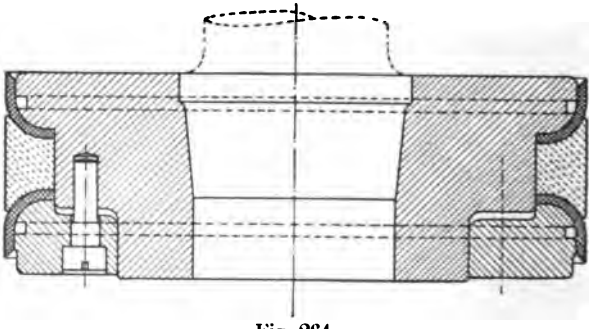


Fig. 264.

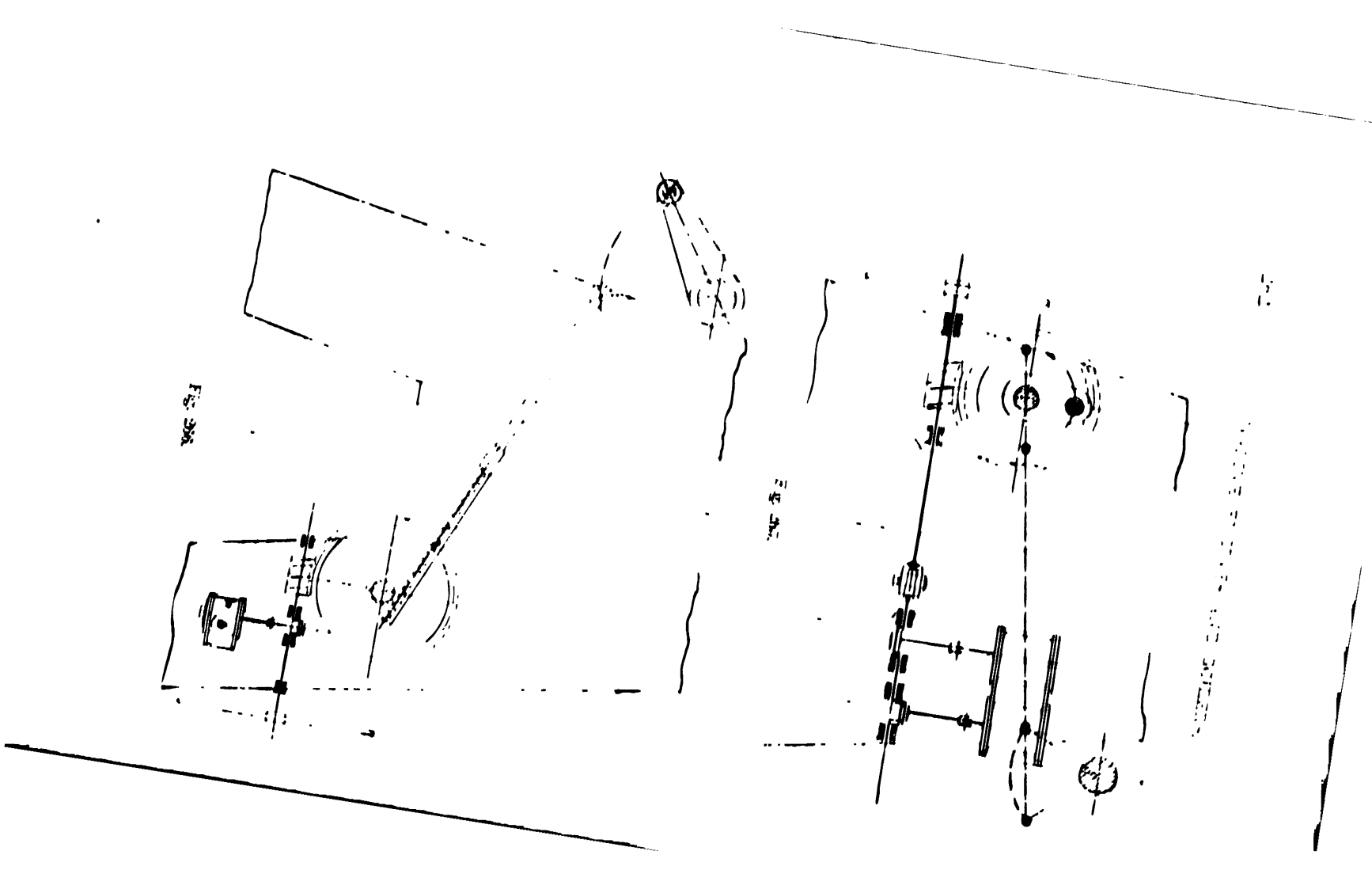
in the same way as an ordinary cast-iron piston. The brake cylinder piston has either the usual metal packing rings, or a double leather bucket packing (see Fig. 264) between which is a turned ring of white metal.

The diameter of the brake cylinder is about 0·6 to 0·7 that of the steam cylinder. The maximum pressure in the brake cylinder is therefore about 2 to 2·8 times greater than the maximum steam pressure in the steam cylinder. The steam cylinder is generally made with a piston rod passing through its upper end only, while the hydraulic or brake cylinder has a rod passing through both covers, and is of the same diameter throughout. This obviates the necessity for an expansion chamber, into which the fluid is forced, and withdrawn, which would be necessary if the piston rod were not carried through, and the volume above and below the piston were different. The reversing engine should be placed as close as possible to the main reversing lever which it operates. The hand gear, for working the reversing gear by hand, should also be brought down to the engine platform with the fewest possible number of joints, levers, and shafts.

The dimensions of the hand gear should be so proportioned that the power required to work the engine is not too great, nor the twisting and bending strains thrown upon the rods so considerable that they bend, or fail to act satisfactorily. The steam for working the engine should be taken both from the main and the auxiliary steam pipes. The exhaust is usually led to the condenser.

§ 144. **All-round Reversing Gear.**—This generally consists of a single or double cylinder steam engine, on the crank shaft of which a worm is fitted driving a worm wheel. The arrangement of a two-cylinder reversing engine of this kind is shown in Fig. 265. A crank on the worm-wheel shaft, or a crank pin fixed direct to the worm wheel, actuates the reversing shaft by means of a connecting rod. In Fig. 266 a diagram is given of an auxiliary reversing engine, which is used in very large marine engines as a stand-by. The reversing gear is actuated by a rack and pinion, the latter being connected to a worm wheel by means of a clutch coupling. An engine drives the worm wheel through a worm.

The reversing of both these small engines can be effected either by a revolving slide valve, as shown at Figs. 267, 268, or by a change-over valve as in the small two-cylinder engines. The revolving or reversible slide valve is so arranged that it acts on one side like an ordinary D valve, and on the other like a so-called "E slide valve," which in its mid position keeps all the ports closed. The valve is turned by means of a hand lever, the valve rod being square in section beyond the valve chest, and also in the slide valve itself. The slide is worked by an eccentric set at an angle of 90° with the crank. If the reversing is effected by a change-over valve, the distributing slide valve is made like an ordinary piston valve, and is operated as in the other case by an eccentric set at 90° to the crank. The change-over valve is



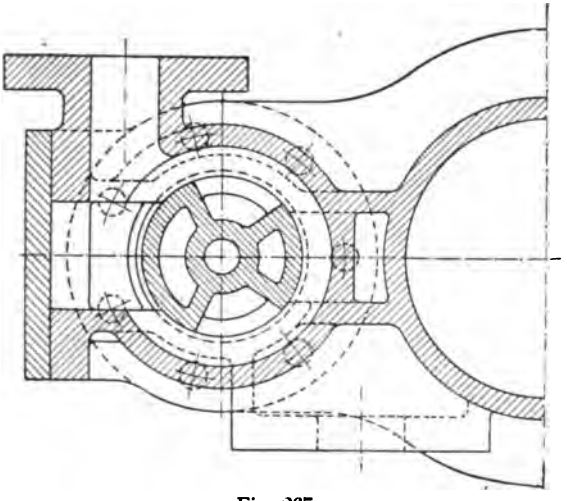


Fig. 267.

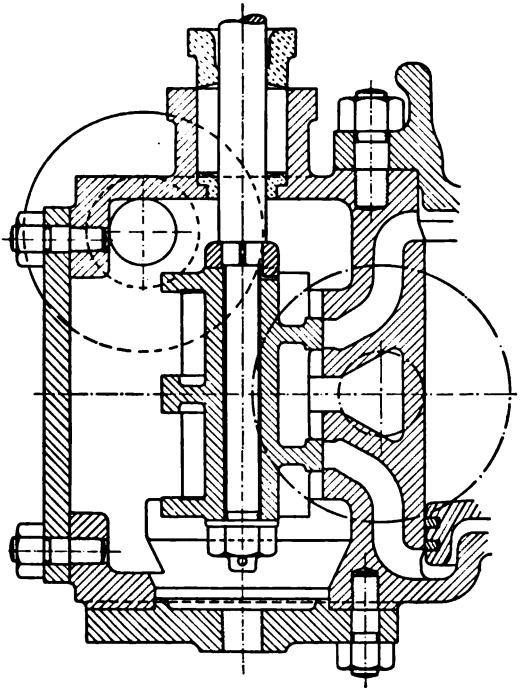


Fig. 268.

in a separate valve chest, and is worked by hand from the platform; it admits steam into and discharges it from the piston valve chest. Steam is thus admitted to the inner or to the outer side of the piston valve, according to the position given to the change-over valve. (In both the above types of engines, the valves have a very small inside and outside lap.)

A flywheel is generally placed on the crank shaft of the reversing engine, which acts at the same time as a hand wheel, to throw over the gear of the main engines by hand. The diameter of this wheel varies from 2 feet 6 inches to 5 feet 3 inches, according to the size of the engine. In the smaller sizes it is made wholly of cast iron; in the larger it has a wrought-iron rim and arms, with a cast-iron boss. Projections on the outside of the rim should be avoided. If of cast iron, the cross section of the rim is elliptical; if of wrought iron, it is generally round, and its diameter from $1\frac{1}{2}$ to 2 inches. In lighter-built ships it consists of a bent tube.

§ 145. **Principal Dimensions of Reversing Engines.**—These, arrived at from the following equations, are as follows:—

(1) $f \times s = c_1 \times \frac{Q \times r}{p}$ for direct-acting reversing engines.

(2) $f \times s = c_2 \times \frac{Q \times r}{p \times n}$ for all-round reversing engines.

Here Q = the maximum load on one main valve rod in pounds.

r = half stroke of main engine eccentric in feet.

p = absolute steam pressure in the main steam pipe in pounds per square inch.

f = area of the steam piston of the reversing engine in square inches. If there are two cylinders, f is the sum of the areas of both pistons.

s = stroke of piston of reversing engine in feet.

n = number of revolutions required in an all-round reversing engine, to reverse the main valve gear from "full ahead" to "full astern."

c_1 and c_2 = constants given in the following table.

TABLE No. 24.

Coefficients for Calculating the Dimensions of Reversing Engines.

c_1	c_2	Type of Main Engine.
2·7	6·7	Single-cylinder engine.
3·8	9·3	Compound engine with two cranks at an angle of 90°
5·4	13·4	Triple „ „ three „ „ 120°
7·6	18·6	Quadruple „ „ four „ „ 90°

In determining the dimensions of reversing engine details, the full boiler pressure must be taken as a basis. The stress on the pins, rods, &c., if they are of wrought iron or steel, should be from 4,250 to 7,000 lb. per square inch; the stress in the teeth of the worm wheels, if of cast iron, from 2,850 to 3,550 lb. per square inch; and from 3,550 to 4,250 lb. per square inch if of bronze or steel, on the assumption that the whole load is taken by two teeth. This corresponds to the twisting moment $\left(\frac{d^3}{10} \cdot k\right)$ in the main reversing shaft. According to the size of the engine the worm wheels have from 20 to 60 teeth, with from $1\frac{3}{8}$ to 3 inch pitch. The worms are either of mild steel forged in one with the shaft, or of bronze, and keyed on to the shaft. It should be noted that a bronze worm works best in a cast-iron or cast-steel worm wheel, and a steel worm in a cast-iron or gunmetal worm wheel. In order to avoid backlash, the teeth of the worm wheel should be milled out by a cutter. If this is done, the worm must be accurately adjusted to the wheel, so that it may be in proper alignment.

The diameter of the pitch circle of the worm is about 1·8 to 2·5t if the worm is of steel; if of bronze and separately cast it is 2·5 to 3·5t, where t denotes the pitch. For convenience in making and fitting, the slope of the worm thread is made straight, whereas the teeth of the worm wheel are slightly hollowed. Length of the worm from 3 to 3·5t. Width of rim of worm wheel 0·6 to 0·8 times the diameter of the pitch circle of the worm. To diminish the wear of the teeth, care must be taken to ensure sufficient lubrication. The best way to secure this is to make the worm or the wheel work in an oil bath.

§ 146. **Turning Gear.**—Engines with a stroke of not more than 14 inches can usually be turned by means of a hand lever. This may be applied either at the forward end of the shaft, or in holes specially bored in the circumference of a coupling flange, or of a small separate wheel fitted especially for that purpose. Larger engines have a worm wheel on the crank shaft, which is turned by a worm and hand lever. If the power required to turn the worm is too great to be worked direct by hand, it is turned through toothed gearing, or by special worm gear and a small steam engine. Instead of having a separate engine, the reversing engine or centrifugal pump engine is often utilised for this purpose.

The turning moment M_1 on the crank shaft required to turn the main engine is—

$$M_1 = C\frac{N}{n}D \text{ ft. lb.}$$

Here N is the I.H.P. of the main engine.

n ,, number of revolutions per minute of the main engine.

D ,, diameter of the crank shaft in feet.

C is a coefficient, which is about 280 for engines of very light build, and about 560 to 670 for heavily built engines.

The required turning moment M_1 may be more accurately determined from the following equation—

$$M_1 = \frac{D}{10}(2G_1 + G_2) \text{ ft. lb.}$$

Here G_1 is the total weight of the pistons, piston rods, crossheads, and connecting rods in pounds.

G_2 is the total weight of the shafting (including the screw) in pounds.

Further, referring to Fig. 269—

z_1 is the number of teeth in the worm wheel A on the crank shaft of the main engine.

z_2 is the number of teeth in the worm.

f is the cylinder area of the turning engine in square inches. (If there are two cylinders of equal size, f is the sum of the areas of both cylinders.)

s is the stroke of the turning engine in feet.

p is the boiler pressure (pounds per square inch) above atmosphere.

i = total reduction in gear from the crank shaft of the main engine to that of the turning engine.

Assuming that the worms s_1 and s_2 are single threaded, then—

$$i = z_1 \times z_2, \text{ and } f \times s = 150\frac{M_1}{i \times p}$$

The total reduction in gear i varies from 1,500 to 4,000, and as the turning engine generally runs at 250 to 400 revolutions per minute, from 4 to 16 minutes are required to turn the main engine through one revolution. The diameter of the pitch circle of the worm wheel on the crank shaft is from 1·2 to 1·6 times the stroke of the main engine.

The materials used for the worm wheel are generally cast iron or cast steel, seldom bronze. The corresponding worm is made of bronze, cast iron, or mild steel, and so arranged that mild steel works upon bronze or cast iron, and bronze upon cast iron or cast steel. Mild steel upon cast steel is not to be recommended, as they do not wear well.

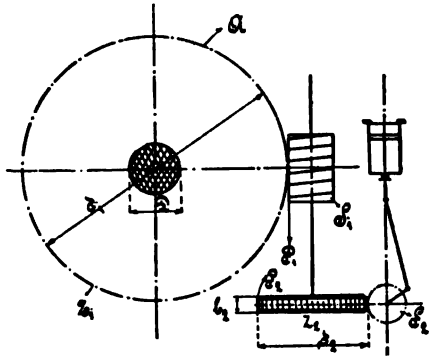


Fig. 269.

If the worm is made of mild steel, it is generally forged in one piece with the spindle; if of cast iron or bronze, it is held on the steel spindle by a taper cotter, or preferably by a key and nut. To enable the turning gear to be thrown out of gear when the engine starts, either the worm s_1 (Fig. 269) must move axially along its shaft, or it must be possible to throw both worm and shaft out of gear from the worm wheel. The latter is the arrangement adopted in the turning gear shown in Figs. 270 to 274. In Figs. 271 to 273 the worm is thrown out of gear by means of a hand wheel, which turns a crank shaft by means of a worm and worm wheel. The crank pin is fitted with a swivelling bearing, which can slide on the worm shaft, and at the same time forms the bottom bearing in which the worm shaft rotates. In the turning gear shown in Fig. 274 the worm is thrown in and out of gear direct by hand. The worm wheel on the main crank shaft should be fixed on

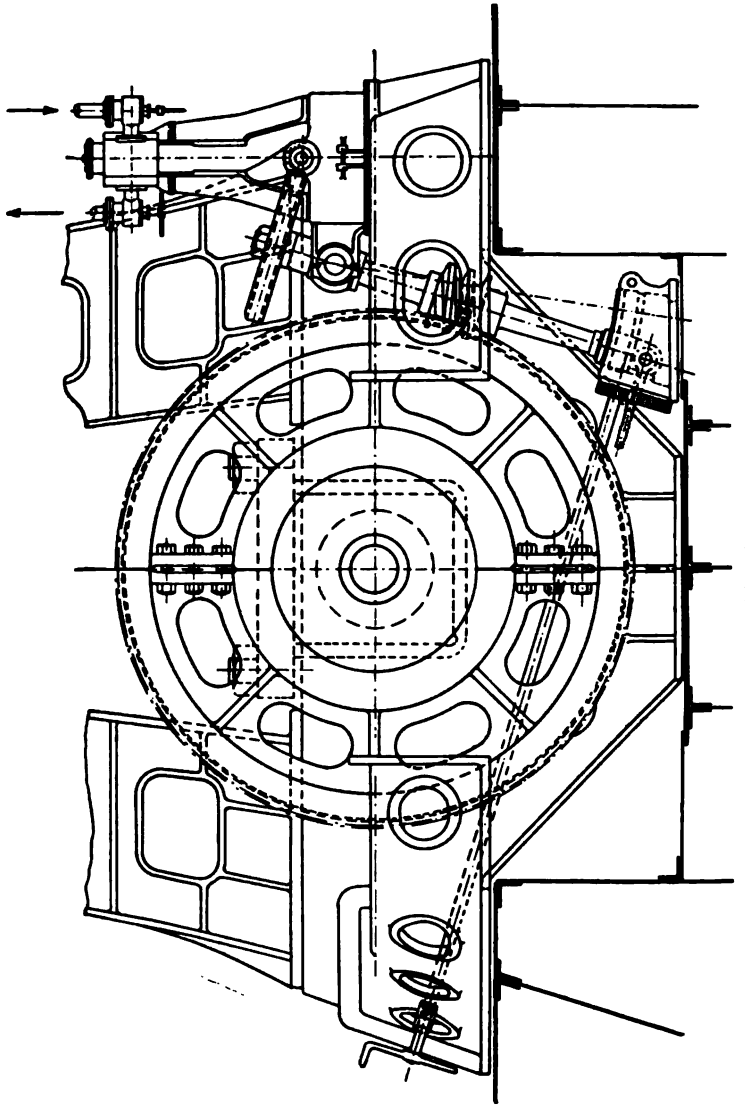


Fig. 270.

a coupling, and to facilitate fitting is usually made in two halves, so that the thrust shaft or after crank can easily be taken out and replaced.

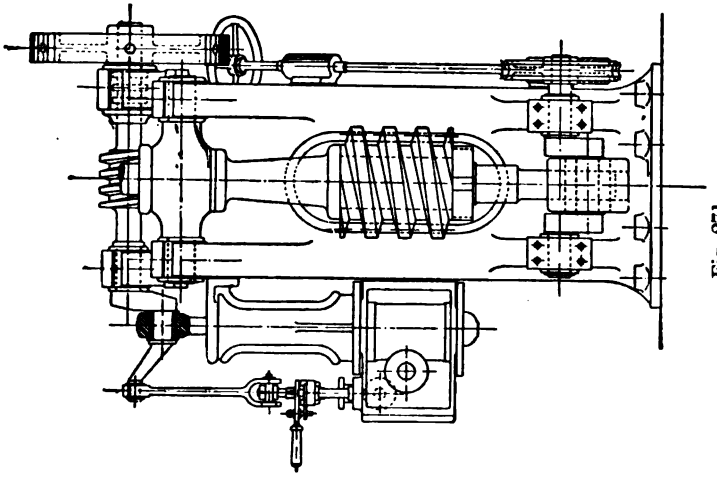


Fig. 271.

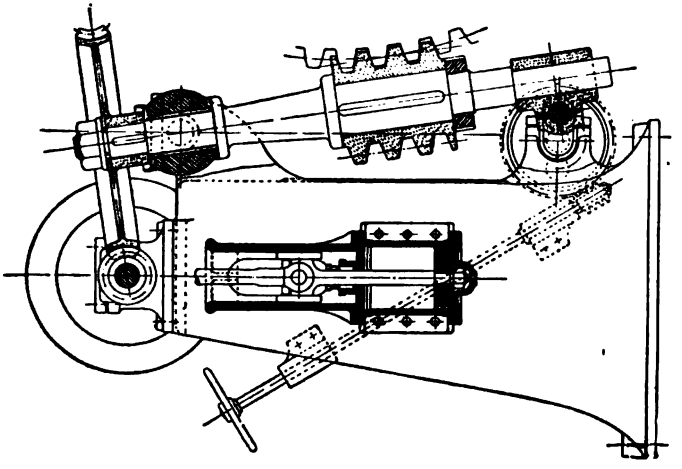


Fig. 272.

§ 147. **Calculation of the Dimensions of the Wheels.**—The following are the symbols employed in regard to Fig. 269 :—

- P_1 Tangential force in the pitch circle of the large worm wheel A in pounds.
- d_1 Diameter of the pitch circle of the large worm wheel A in feet.
- t_1 Pitch of the large worm wheel A in inches.
- b_1 Width of the large worm wheel A in inches.
- P_2 Tangential force in the pitch circle of the small worm wheel in pounds.

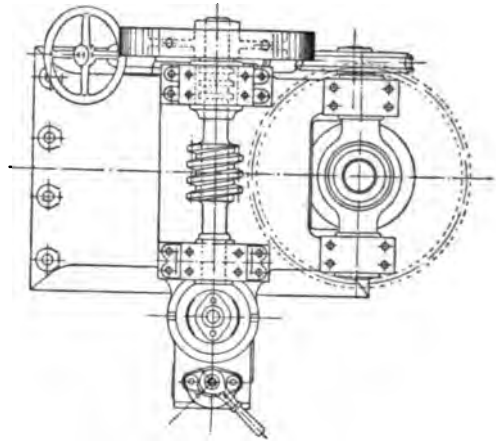


Fig. 273.

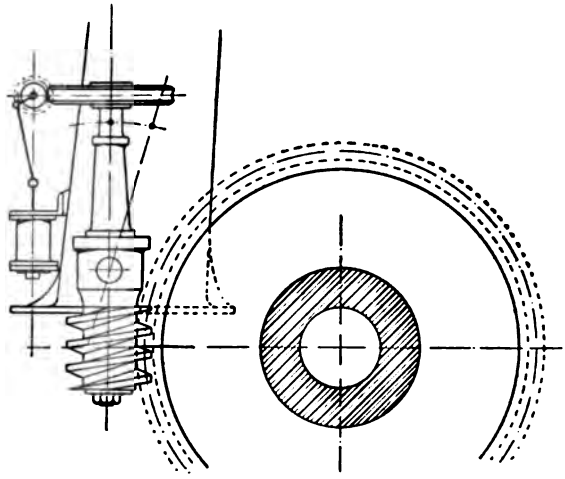


Fig. 274.

d_2 Diameter of the pitch circle of the small worm wheel in feet.
t_2 Pitch of the small worm wheel in inches.
b_2 Width of the small worm wheel in inches.
c_1 and c_2 are constants.
z_1 Number of teeth in the large worm wheel A.
z_2 Number of teeth in the small worm wheel.

Then $P_1 = 2\frac{M_1}{d_1}$; $b_1 \times t_1 = \frac{P_1}{c_1}$; $P_2 = \frac{6M_1}{z_1 d_2}$ (with an efficiency = $\frac{1}{3}$ and a single thread worm gear s); $b_2 t_2 = \frac{P_2}{c_2}$.

The constants c_1 and c_2 may be used on the assumption that the strain is taken by two teeth simultaneously, and in the accompanying Fig. 275, $f = 0{\cdot}6t$; $l = 0{\cdot}65t$. The constants c_1 and c_2 are—

For cast iron or bronze $c_1 = 640$
„ „ $c_2 = 355$
For cast steel $c_1 = 895$
„ „ $c_2 = 500$

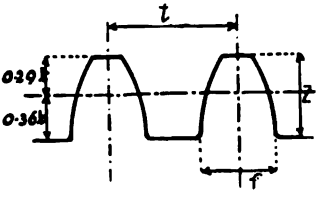


Fig. 275.

The stress in the teeth, if made of cast iron or bronze, is taken at 3,500 lb. per square inch; if of cast steel, at 5,000 lb. per square inch. As the thread of the worm wears out much quicker than the teeth of the worm wheel, it is advisable, especially if the worm is made of bronze, to make the teeth of the worm considerably thicker than the teeth on the wheel. If the worm wheels are milled by a cutter, the clearance between the thread of the worm and the teeth of the worm wheel need only be from $0{\cdot}05$ to $0{\cdot}03t$.

Thickness of the rim of the worm wheel inside the teeth is about $0{\cdot}05t$; thickness of the arms and ribs about $0{\cdot}4$ to $0{\cdot}45t$.

Width of the rim of the worm wheel and of the teeth is as follows:—

$$b_1 = 2{\cdot}4 \text{ to } 2{\cdot}8t_1;\ b_2 = 2{\cdot}4 \text{ to } 2{\cdot}8t_2.$$

The boss is generally as wide as or rather wider than the rim of the wheel.

Radial thickness of boss, if the wheel is of cast iron, is about $1{\cdot}2$ to $1{\cdot}5t$.
„ „ „ cast steel, „ $0{\cdot}75$ to $0{\cdot}95t$.

The diameter of the pitch circle of the worm is—

From $1{\cdot}5$ to $2{\cdot}5t$ if the worm and shaft are forged in one piece.
„ $2{\cdot}5$ „ $3{\cdot}5t$ „ is fixed on separately.

Radial thickness of the metal below the thread about $0{\cdot}5$ to $0{\cdot}6t$.
Length of worm should be at least from 3 to $3{\cdot}5t$.

Condensers.

1. Surface Condensers.

§ 148. **General Remarks.**—Modern vessels are invariably fitted with surface condensers. The weight, cost, and space occupied are considerable, but the recovery of the feed water outweighs these disadvantages. The condenser tubes are generally horizontal or slightly inclined, and only in very exceptional cases vertical. The steam to be condensed generally enters from above, and circulates outside the condenser tubes, and care must be taken that as far as possible it is equally distributed over the whole length and breadth of the tubes. To effect this a sheet of galvanised iron pierced with holes is fixed immediately beneath the steam inlet (see Fig. 279), and this serves at the same time to prevent the steam impinging directly on the tubes. These baffle plates must be stiff enough to stand the current of steam striking against them without deflection, as otherwise they might injure the uppermost rows of tubes. In the upper part of the condenser some of the tubes are sometimes omitted in order to facilitate the passage of the steam into the centre of the nest of tubes. (See Figs. 276 and 281.)

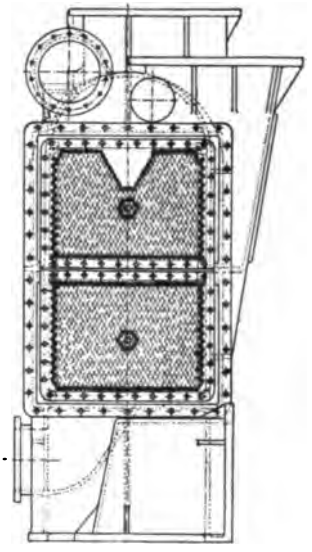


Fig. 276.

The circulating water is generally led twice through the tubes, that is, it is led forward through half of them, and back through the other half. Whether the water is admitted at the bottom and discharged at the top, or *vice versâ*, does not materially affect the vacuum produced. Surface condensers in which the circulating water passes outside the tubes, and the steam to be condensed passes inside them, are hardly ever made in modern practice. If the circulating water is admitted at the bottom, the temperature of the condensed steam will be lower than if the water

is passed through the condenser in the reverse direction, because in the former case the condensed steam comes in contact with the coldest condenser tubes last of all.

The air-pump suction is connected to the lowest part of the condenser; if the air pump is separate from it, the two must be connected by a suction pipe of sufficient diameter. For the size of the air-pump suction pipe see page 286. For dimensions of inlet and outlet for the circulating water see pages 294 and 298. If a plunger pump is used for the circulating water, it is desirable to form an air chamber on the cover of the condenser, on the side at which the circulating water enters. By this means water-hammer and the resulting injurious strains to the condenser are avoided. When placing the condenser in the ship, care must be taken that sufficient space is left at one end to enable the tubes to be withdrawn and replaced. This space should not be encroached on by any parts of the engine, piping, &c. Sometimes a hole, over which a plate is fitted, is left in the nearest bulkhead, through which the tubes can be drawn.

§ 149. **Cooling Surface.**—The surface required per I.H.P. can be taken at the following values—

1·5 to 1·7 square feet in compound engines.
1 „ 1·5 „ triple engines.

The external surface of the cooling tubes is reckoned as the cooling surface, and the tubes are arranged as shown in Fig. 277. The following

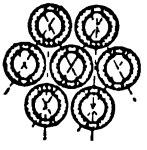


Fig. 277.

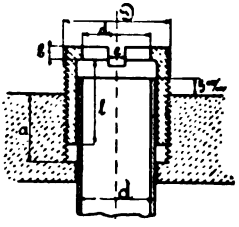


Fig. 278.

Table No. 25 gives the amount of cooling surface per linear foot of tube and per cubic foot of nest of tubes, for different diameters and pitches of tubes.

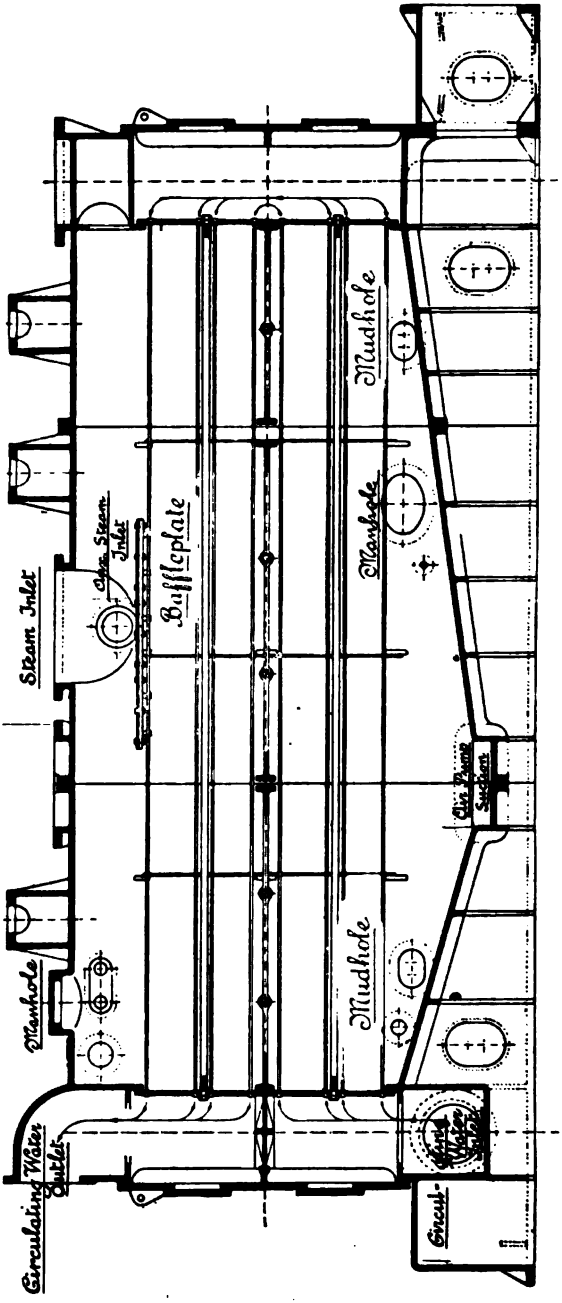

Circulating Water Outlet
Manhole
Steam Inlet
Blow Steam Inlet
Baffleplate
Mudhole
Manhole
Mudhole
Air Pump Suction
Circul. Water Inlet
Circul.

Fig. 279.

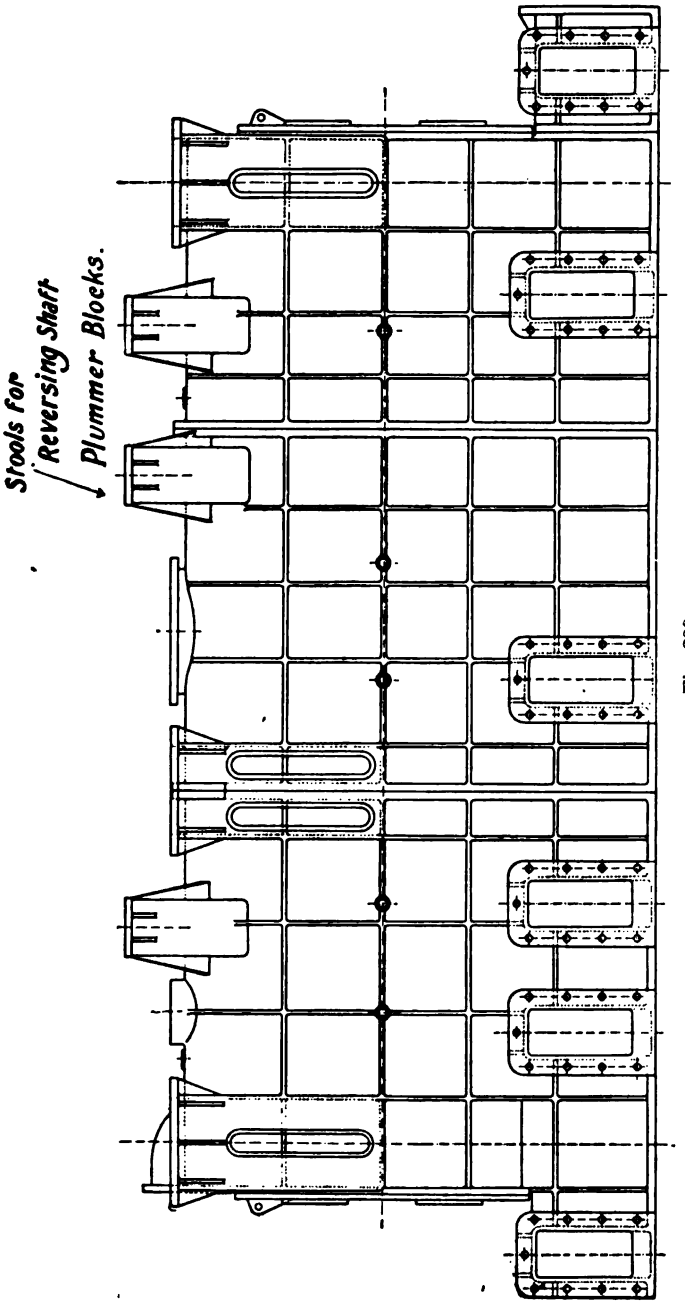


Fig. 280.

TABLE No. 25.

Number in a square foot.	Pitch of Tubes t in inches.	External Diameter of Tubes in inches.					
		$\frac{1}{2}$ inch.	$\frac{5}{8}$ inch.	$\frac{3}{4}$ inch.	$\frac{7}{8}$ inch.	1 inch.	
		sq. ft.	sq. ft.	sq. ft.	sq. ft.		
172	1	22·515	28·139				Cooling surface (in sq. ft.) per cubic foot of nest of tubes.
150	$1\frac{1}{16}$	19·635	24·54	29·445			
137	$1\frac{1}{8}$	17·933	22·413	26·893	31·386		
128	$1\frac{5}{32}$	16·855	20·94	25·126	29·325		
121	$1\frac{3}{16}$		19·795	23·752	27·721		
116	$1\frac{7}{32}$			22·77	26·575		
110	$1\frac{1}{4}$				25·201	28·798	
99	$1\frac{5}{16}$				22·68	25·918	
Cooling surface in square feet per foot run of 1 tube		·1309	·1636	·1963	·2291	·2618	sq. ft.

NOTE.—Table No. 26 has been incorporated in the above table.

§ 150. **Tubes and Tube Plates.**—The *tubes* are of solid-drawn brass, containing about 60 to 70 °/₀ copper and 30 to 40 °/₀ zinc (an alloy of 70 % copper, 29 % zinc, and 1 % tin has been found to answer well), and are often tinned inside as well as out. Their external diameter generally varies between $\frac{5}{8}$ and $\frac{7}{8}$ inch, and their thickness between 19 B.W.G. and 16 B.W.G., the figure generally used being 18 B.W.G. (0·049 inch). The maximum length of the tube between the tube plates is about 20 feet. If the length is more than 80 to 100 times the external diameter of the tubes, they must be supported by special intermediate plates of sheet brass, from $\frac{5}{16}$ to $\frac{3}{8}$ inch thick. The pitch of the tubes (t), *i.e.*, the distance of the centres of the tubes from each other, is not less than $t = d + 0{\cdot}35$ inch, d being the external diameter of the tube. Table No. 25 gives the cooling surface per foot run of different tubes, as well as the number of tubes in a square foot of tube plate, for different pitches of tube.

Tube plates are now seldom made of cast bronze, but generally of rolled brass or "Muntz" metal (60 % copper, 40 % zinc), from $\frac{7}{8}$ to 1 inch thick. The packing and securing of the tubes in the tube plates is usually done by means of small stuffing boxes and a screwed ferrule or gland, packed with soft cord, as shown in Fig. 278. The ordinary dimensions of these stuffing boxes (see Fig. 278) are—

$$a = 0{\cdot}85d,\ \text{D} = d + {\cdot}25 \text{ inch},\ d_1 = d - {\cdot}094 \text{ inch},\ l = 0{\cdot}95d,$$
$$b = \tfrac{3}{16} \text{ to } \tfrac{7}{32} \text{ inch},\ e = \tfrac{7}{32} \text{ to } \tfrac{1}{4} \text{ inch}.$$

The number of threads to the inch is about 16 to 18. When the tube plates are very large, they must be braced to one another by tie rods to ensure their rigidity, the stays being spaced about 15 to 20 inches apart. (See Figs. 276 and 279, 281 and 282.) These stays are generally thick brass tubes, accurately fitting between the end plates, and secured from the outside by a set screw passing through the plates, and screwed into the tapped bore of the tube.

§ 151. **Condenser Shell.**—This is either cast in one with the engine frame, or made separately. In the former case it is of cast iron,

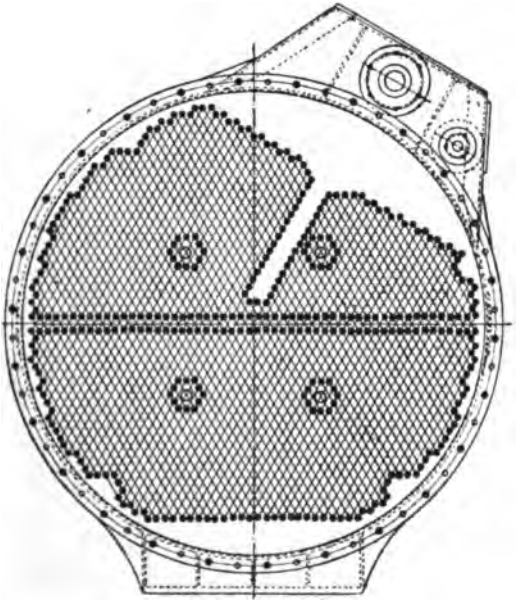
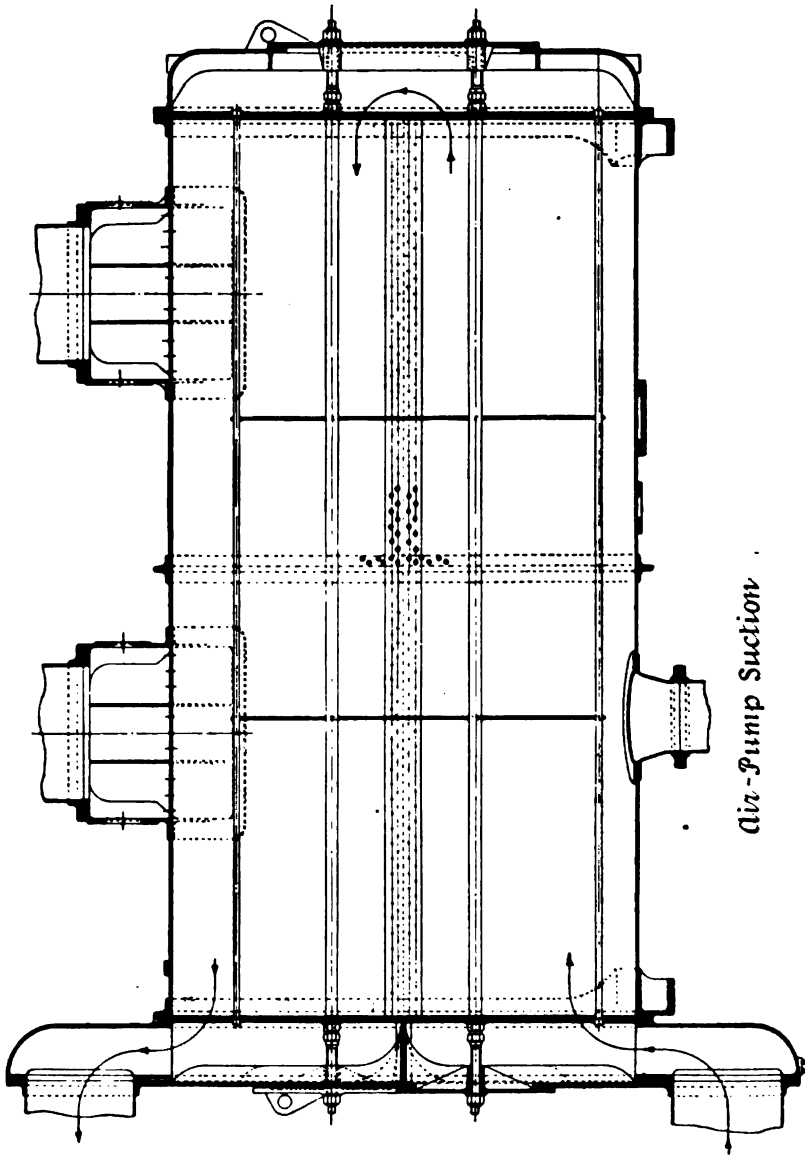


Fig. 281.

and rectangular in shape. With separate condensers the form is generally cylindrical, although in warships they are often oval, the flattened surfaces being stayed. They are made of cast iron, cast gunmetal or bronze, galvanised steel, copper, or brass plates, and tinned inside. If made in one with the framing, the bottom of the condenser is so shaped that the condensed steam can flow easily to the air-pump suction pipe. (See Fig. 279.) With the same object, separate condensers are usually slightly inclined towards the end connected to the air pump.



Air-Pump Suction.

Fig. 282.

The thickness s of cylindrical condenser shells is—

$$s = \frac{d}{180} + \cdot 393 \text{ inch for cast iron.}$$

$$s = \frac{d}{400} + \cdot 196 \quad \text{,, wrought iron or mild steel.}$$

$$s = \frac{d}{300} + \cdot 196 \quad \text{,, cast bronze or gunmetal.}$$

$$s = \frac{d}{350} + \cdot 039 \quad \text{,, rolled copper or brass.}$$

To stiffen the shell and assist in maintaining its form, a few circumferential ribs are fitted. If the condensers are of copper, brass, or iron plates, these ribs are generally T-shaped in section, and riveted on.

The thickness of condenser shells, cast in one with the engine framing, and with flat walls, may be taken from Table No. 27. The flat walls are stiffened by ribs placed at a distance of about $20s$ apart, the height of the ribs being about $4s$. In large condensers of this kind the two vertical longitudinal walls are also sometimes braced together by stay rods passing between the two groups of condenser tubes (Figs. 279 and 280).

TABLE No. 27.

Cooling Surface of Condenser, in square feet.	Thickness s of Condenser Shell having Flat Walls, in inches.
Above 5400	1·181
From 3200 to 5400	1·106
,, 2150 ,, 3200	0·984
,, 1600 ,, 2150	0·866
,, 1000 ,, 1600	0·787
,, 540 ,, 1000	0·708
Below 540	0·511 to 0·708

In cylindrical condensers the water chambers at each end are made of cast iron, bronze, or of copper plates. They have to be entirely removed to reach and overhaul the tube plates and stuffing boxes, unless they happen to be fitted with a large inspection door which can be easily removed. The ends of large condensers are therefore always fitted with manholes and manhole doors, to allow the tube stuffing boxes to be more easily examined. The tube plates are fixed to the condenser shell and ends in the manner shown in Figs. 279 and 283.

To test the condenser and packing for leakage, they are subjected to a water pressure of from 30 to 40 lb. per square inch; first the shells only, with the covers but without the tubes, are tested, and then the steam space, after the tubes have been put in, to make sure that the tubes, stuffing boxes, and shell joints are tight.

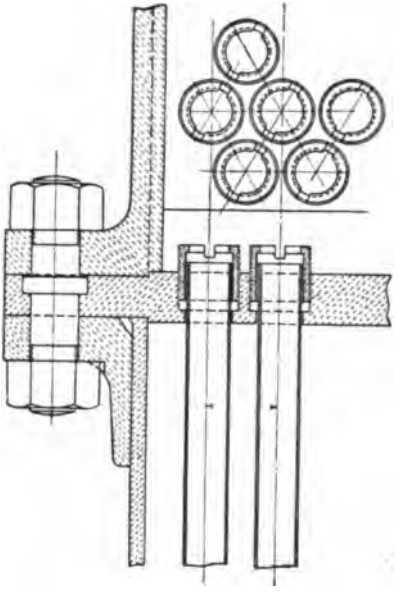


Fig. 283.

§ 152. **Fittings and Connections.**—Besides the opening for the main steam exhaust, inlet and outlet of the circulating water, and the air-pump suction, the following connections are generally provided:—Auxiliary engine exhaust, electric light engine exhaust, reversing engine exhaust, and the drains from the steam cylinders and valve chests.

On the top of the condenser shell a manhole is usually fixed, and at the bottom one or more mud-holes. The bottom of the condenser has also a valve and pipe connection from $\frac{1}{2}$ to $\frac{7}{8}$ inch internal diameter, to allow of its being boiled out with steam. At a suitable place there is also a cock $\frac{3}{8}$ to 1 inch internal diameter, through which the make-up feed water is added. In modern practice this pipe is usually connected to the reserve feed tank only. A soda cock, about $\frac{1}{2}$ inch internal diameter, fitted with funnel or hose, is also provided. At the lowest point in the bottom of the condenser there is another valve from $1\frac{1}{2}$ to 3 inches internal diameter, according to the size of the condenser, through which, if the air pump is not working, the condenser can be kept clear by a steam pump. Drain cocks are placed as required in the lower part of the condenser, and generally also a special valve, through which the condenser can be filled direct from the sea. To protect the inside of the condenser, where it is in contact with the sea water, against corrosion, zinc slabs are often fitted.

2. Jet Condensers.

§ 153. **Jet Condensers.**—These are now used only in river

steamers, and consist of an arrangement by which a jet of water taken from the river is injected into the exhaust pipe, or into the hollow column of the engine through which the exhaust is carried. In case of need the injection water can also be drawn through a special cock from the bilge (so-called auxiliary injection). The quantity of injection water is regulated by a cock or valve placed in the admission pipe, which can be easily manipulated by means of levers from the engine platform.

The area of the injection pipe is as follows:—

·025 to ·031 square inch per I.H.P. in compound engines.
·0186 „ ·025 „ „ triple-expansion engines.

The diameter of the holes in the rose of the injection pipe is about $\frac{1}{8}$ to $\frac{3}{16}$ inch, and their total area is about 0·5 the area of the injection pipe. If there is any danger of the small holes getting stopped up, the water is admitted over a weir. An automatic valve is placed at the bottom of the condenser, which is held shut by the pressure of the atmosphere outside. The object of this valve is to discharge the water automatically from the condenser when the engine is not running.

PART II.

PUMPS.

Air Pumps.

§ 154. **General Remarks.**—The air pump is generally fixed to the main engine, and driven either direct or by levers from one of the crossheads, the latter being the most usual arrangement. In modern practice the air pump, especially in large engines, is often entirely independent of the main engines. It then forms a separate engine, and is sometimes combined with the circulating pump, as in the Blake, Weir, Worthington, and other systems. In the mercantile marine, if the air pump is to be driven from the engine, there is generally only one pump, while in warships two are frequently provided for each main engine. In the latter case the suction and delivery pipes of the pumps should have suitable screw-down or sluice valves, so that each air pump may be worked independently of the other, if necessary.

Air-pump barrels are generally vertical, and the pump should be placed as low as practicable, but the foot valves should at least be at the same level as the bottom of the condenser. The amount of clearance space in the air pump has not a very serious effect on the vacuum produced, because it is filled with water.

§ 155. **The principal dimensions** of single-acting air pumps are determined from the equation—

$$f \times s = C\frac{\text{I.H.P.}}{n}$$

Here f = sectional area of the pump cylinder in square inches.

s = stroke of the pump cylinder in inches.

I.H.P. = indicated horse power of the main engine.

n = number of double strokes of the air pump per minute.

C = constant, equal to volume delivered by the air pump per I.H.P. per minute.

The coefficient C = 86 to 111 in surface condensers of triple or quadruple expansion engines, with separately driven air pumps. (See § 161.)

C = 185 to 245 in surface condensers of triple or quadruple expansion engines, the air pump being driven by the main engine.

C = 300 to 365 in surface condensers of compound engines, the air pump being driven by the main engine.

If jet condensing is used as well as surface condensing, or if the former alone is used, c = 610 to 730. If, instead of one air pump, two pumps are fixed to and driven by the main engine, the volume swept through per stroke in each is about 0·6 of that given above.

Piston Speed.—In cargo steamers the mean speed of the air-pump piston varies from 200 to 350 feet per minute, and in warships from 300 to 500 feet per minute.

§ 156. **Air-pump Valves.**—Air pumps have, as a rule, suction, plunger, and delivery valves, although, generally speaking, either suction and plunger valves, or plunger and delivery valves should be sufficient.

In the Edwards' patent air pump, shown in Fig. 286, there are delivery valves only. Here the water collecting below the bucket is, as the bucket descends, forced through openings at the side of the barrel on to the top of the bucket. As the bucket rises, the openings at the side are closed, and water and air are forced through the delivery valves into the hot-well tank. The net sectional area of the valve openings should be such, that the mean velocity of water through them does not exceed 13 to 16·5 feet per second. The radial speed of discharge at the circumferential ports or openings should not be more than 20 to 33 feet per second.

The valves are of various types. Those in most frequent use are flexible rubber flap valves (Fig. 285) from $\frac{1}{2}$ to $\frac{5}{8}$ inch thick. "Beldam" valves (Fig. 284) are also often used, consisting of a corrugated brass plate about $\frac{1}{16}$ inch thick, and a valve guard corresponding in shape. "Kinghorn" valves (Fig. 287) consist of three loose plates of sheet brass from about $\frac{1}{32}$ to about $\frac{1}{16}$ inch thick, placed one over the other, the two lower having holes about $\frac{1}{8}$ to $\frac{3}{16}$ inch diameter drilled in them at different radii. Ordinary rubber flap valves are fixed at the centre, and a little "play" is allowed, the lift at the circumference due to their elasticity being about $\frac{5}{8}$ to $\frac{7}{8}$ inch. Beldam and Kinghorn valves sit loose on the valve spindle, and work up and down on it, the lift being only from about $\frac{1}{8}$ to $\frac{5}{16}$ inch. The greater the number of strokes of the bucket per minute the smaller should be the lift of the valves. The clear openings between the ribs of the valve seatings should not be more than about $1\frac{3}{16} \times 1\frac{3}{16}$ inch, while the width of the ribs in the guards should be at least $\frac{3}{16}$ inch. The valve seats, guards, and spindles are made of gunmetal. It is best to screw the spindles into the seat, and rivet them over on the under side. The valve guards are screwed on, and secured by lock nuts and split pins.

§ 157. **Suction and Delivery Pipes.**—With surface condensers the sectional area of the air-pump suction pipe, or the connection to the

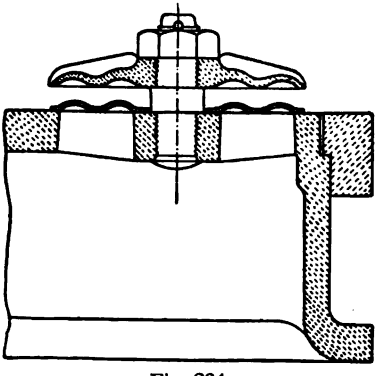


Fig. 284.

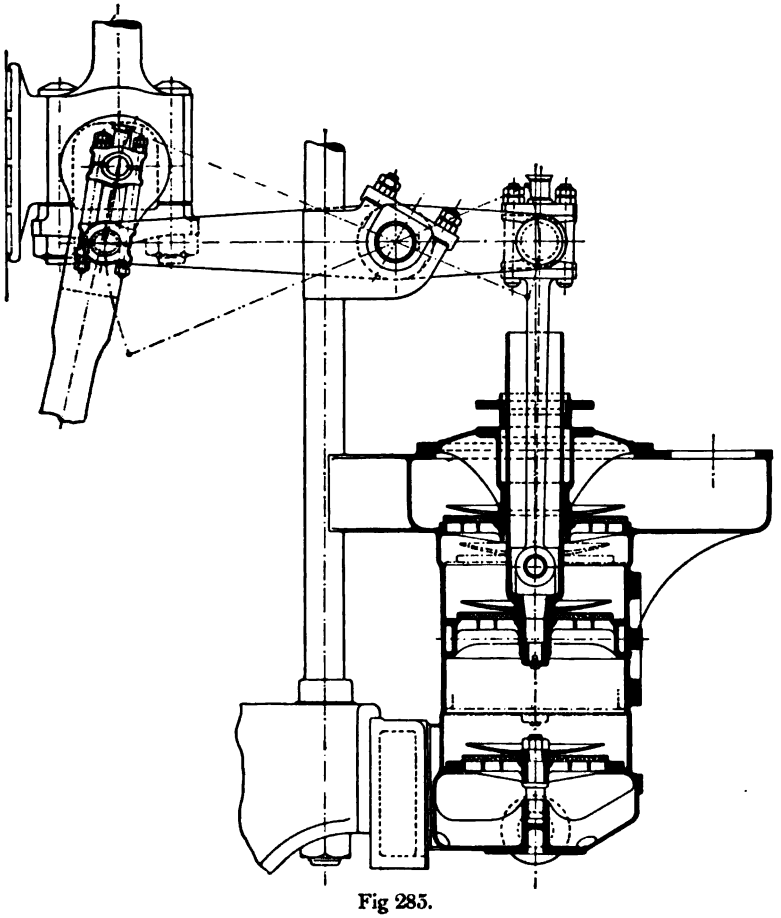


Fig 285.

condenser, should be such that the mean velocity of air and water in the pipe, calculated from the volume of the effective strokes of the air pump, is from 10 to 16 feet per second. The velocity in the delivery pipe should be about 26 to 33 feet, sometimes it may even be as much as 50 feet, per second, and the sectional area of the pipe must be determined accordingly. With jet condensers the velocities in the suction and delivery pipes should only be about 50 % of the velocities for surface condensers.

When it is not required to use a surface condenser on emergency as a jet condenser, it has been usual of late years not to fit an overboard

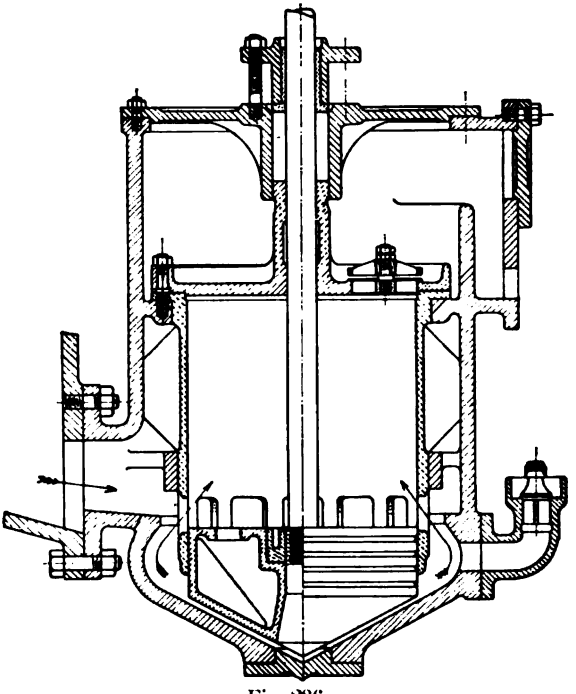


Fig. 286.

air-pump discharge valve, but to leave the end of the delivery pipe open, and let it discharge the air direct into the engine-room, while the water delivered by the air pump either flows over into a hot well tank, or is sucked by the boiler feed pump direct from the delivery chamber of the pump. When an overboard discharge valve is fitted on the discharge pipe, a small branch air pipe is taken from underneath the valve, and is made to discharge either as high as possible above the water line, or straight into the engine-room. Through this pipe the air discharged by the pump can escape, in case the large discharge

valve has to be kept closed, owing to the rolling of the vessel. Its diameter is about 0·2 to 0·3 the diameter of the delivery pipe. Close to the delivery chamber of the air pump an air vessel is placed, the

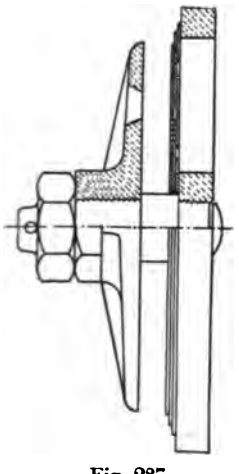


Fig. 287.

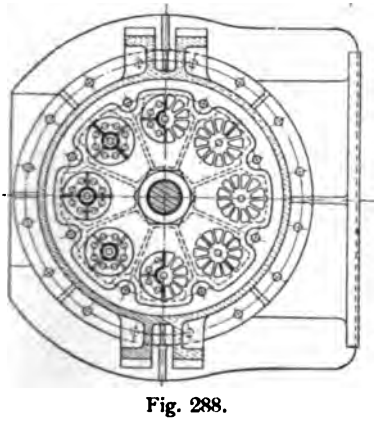


Fig. 288.

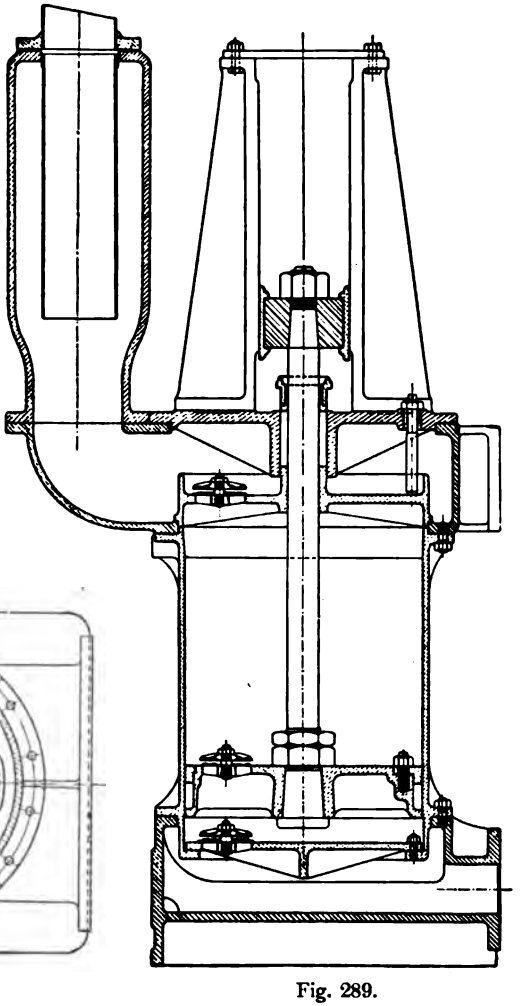


Fig. 289.

cubic contents of which are about 0·3 to 0·5 that of the pump cylinder. Sometimes the delivery chamber of the air pump forms practically the tank from which the feed pumps suck, and is known as the "Hot Well."

§ 158. **Pump Body.**—In merchant vessels the body of the air pump

is made of cast iron, the pump barrel itself being of gunmetal in the form of a liner. In larger pumps a solid gunmetal barrel is sometimes bolted by flanges between cast-iron suction and delivery valve chests. (See Fig. 289.) In warships the pump barrel and suction and delivery valve chests are all of gunmetal, and generally cast in one piece. (See Figs. 285, 290, 292.) To make the bucket and foot valves in large air pumps easily accessible, the barrel of the pump is made with a round manhole at the side, from 10 to 12 inches in diameter, the cover of which must, of course, form part of the cylinder wall. (See Figs. 285 and 290.) In the same way hand holes with covers are made in the sides of the delivery valve chest, so that these valves also can be easily reached.

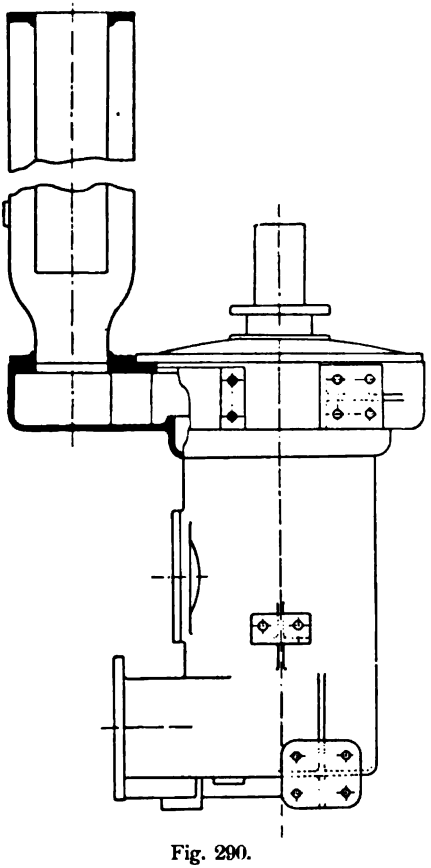


Fig. 290.

The thickness of the walls s of the gunmetal barrel of the pump is $s = 0{\cdot}015D + {\cdot}25$ inch, D being the diameter of the air pump in inches. The thickness of the walls of a cast-iron pump body is $s_1 = 0{\cdot}015D + {\cdot}39$ inch. The thickness of metal in the suction and delivery valve chests of the air pump are, in warships, from $\frac{1}{16}$ to $\frac{1}{8}$ inch thinner than the pump barrel. The foot-valve seating is fixed to the lower part of the pump either by several set screws round the rim, or by a strong bolt through the centre. (Compare Figs. 285 and 292.) The head or delivery valve seating is generally held and made tight against the pump rod by the main stuffing-box in the centre, and at the rim either by set screws or by a few long bolts screwed right through from the upper valve-chest cover. The thickness s_2 of the valve seat is $s_2 = 1{\cdot}1$ to $1{\cdot}2s$, strengthened by radial ribs underneath.

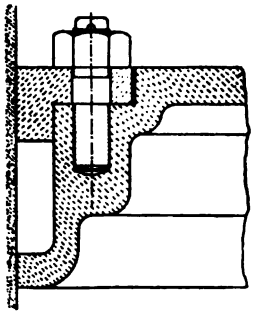


Fig. 291.

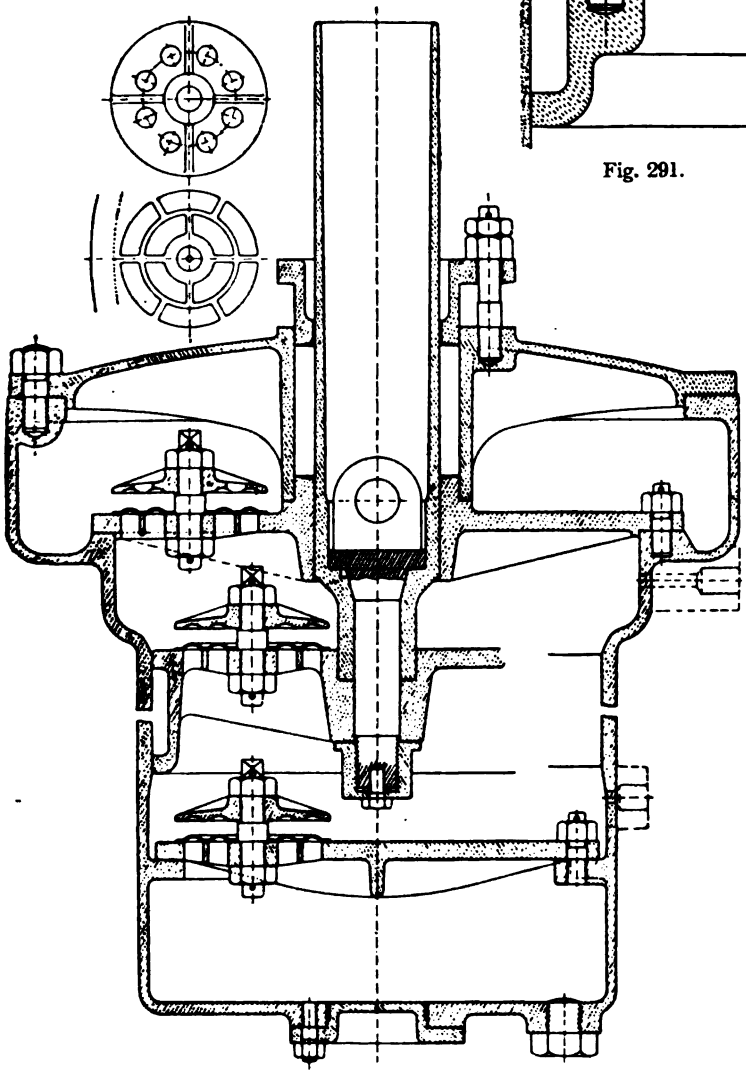


Fig. 292.

§ 159. **The Pump Bucket** is made of gunmetal, and the thickness s_3 of the flat portions of the piston is $s_3 = 0{\cdot}02D + 0{\cdot}23$ inch. It is stiffened with radial ribs, about $0{\cdot}5s_3$ thick, placed between the openings for the valve seats. The bucket is usually packed with cotton or hemp, which in small buckets is wrapped round a deep rectangular groove on the outside of the bucket or piston, while in the larger sizes it is packed into a similar space, and pressed down by a junk ring. (See Figs. 284 and 291.)

The ring is not intended to adjust the packing, and is screwed down firmly against the piston by means of studs, so that it cannot work loose. The depth of the packing space h = about $0{\cdot}1D + 1{\cdot}38$ inches. The thickness of the packing space (or the thickness of the packing) = $\frac{h}{4}$ to $\frac{h}{5}$.

§ 160. **The Pump Rod** is made either wholly of phosphor bronze, or of steel, in which case it is fitted with a bronze sleeve. It is fitted into the bucket and crosshead by means of coned ends with nuts, in the same way as an ordinary piston rod. If the rod is of steel, the nut is generally a cap nut, to prevent the water reaching the thread. It is advisable in large pumps to provide a boss in the centre of the suction-valve seat, to support the pump-rod nut, and to prevent the plunger striking against the valves, when detaching the pump rod from its crosshead. If from want of space it is impossible to have a guide and connecting rod above the air pump, the bucket is made as a trunk piston, the connecting rod being attached to a pin at the lower end of the trunk. (See Figs. 285 and 292.)

The diameter of the plain part of the air-pump rod is about $d = 0{\cdot}12D + {\cdot}5$ inch, and the diameter at the bottom of the thread at either end is $d_1 = 0{\cdot}085D$ to $0{\cdot}1D$.

§ 161. **Separately Driven Air Pumps.**—These pumps, which are entirely separate from the main engine, are made by the Blake & Knowles Steam Pump Co., G. & J. Weir of Glasgow, the Worthington Pump Co., and others, and have been much used of late years. Number of double strokes per minute = 12 to 25.

Fig. 294 shows a Blake-Knowles two-cylinder air pump. The two air pumps are single-acting, and each is driven by a separate steam cylinder, the piston rod of which is coupled direct to that of the air pump. To equalise the work, the two piston rods are further interconnected by a beam and connecting links, and therefore move in opposite directions at the same time. The valve gear of the steam cylinders is worked by levers from the beam shaft.

Fig. 293 shows a similar air pump, as made by Messrs Weir. The air pump is often combined with the circulating pump, and both are

worked at the same time by one steam piston. The Blake-Knowles Co., Worthington Co., and others make pumps of this type. In the Blake-Knowles combined pumps the two pumps and the steam cylinder are all placed in a line, on the same axis, the steam cylinder being in the middle. The Worthington pumps are made on the Duplex system, with the two sets of cylinders side by side.

The Blake & Knowles Steam Pump Co. have lately constructed vertical single-cylinder double-acting air pumps, in which, as in the Edwards pump, the pump cylinders are only fitted with delivery valves. (See *Schiffbau*, 1903, p. 951.) Two such pumps, with cylinders of different diameter, can be coupled together through the exhaust of the small cylinder, and be worked as compound pumps.

Circulating Pumps.

§ 162. **General Remarks.**—To provide the cooling water for the surface condensers, either reciprocating or centrifugal pumps are used. Reciprocating circulating pumps are usually worked from the main engine, and driven, together with the air pump, bilge and feed water pumps, by means of a rocking lever. Sometimes, however, the circulating pump is separate from the main engine, and is then usually combined with an independent air pump, as in the Blake, Worthington, and similar systems.

The *quantity of cooling water required* per I.H.P. per hour is: For triple or quadruple expansion engines, about 440 to 550 lb.; for compound engines, about 550 to 650 lb.

Reciprocating Circulating Pumps.

§ 163. **General Remarks.**—This class of pump may be made either single or double acting, and is generally driven, in conjunction with the air and feed pumps, by a beam from one of the crossheads of the main engine. The stroke of the pump is usually half that of the main engines, and the piston speed may reach 490 feet per minute.

The *principal dimensions* of a double-acting circulating pump may be arrived at as follows:—

$$f \times s = \frac{Q \times \text{I.H.P.}}{100 \times n}$$

and those of a single-acting pump—

$$f \times s = \frac{Q \times \text{I.H.P.}}{50 \times n}$$

where f = area of the pump piston in square feet.

s = stroke „ „ feet.

n = number of double strokes per minute.

Q = quantity of cooling water per I.H.P. per hour in cubic feet.

I.H.P. = the indicated horse power of the main engine.

The quantity of cooling water required per I.H.P. per hour is—

8·8 to 10·6 cubic feet in compound engines.

7 „ 8·8 „ triple and quadruple expansion engines.

The above equations allow an efficiency of 83·3 % for the pump itself.

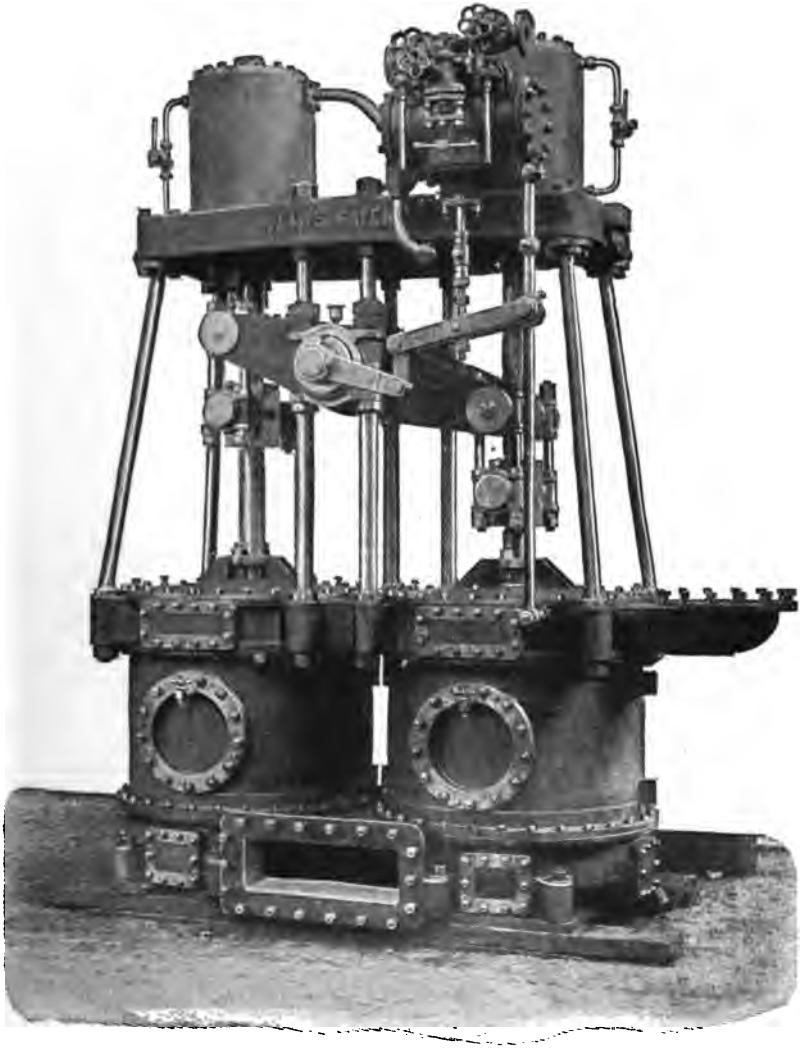


Fig. 293.

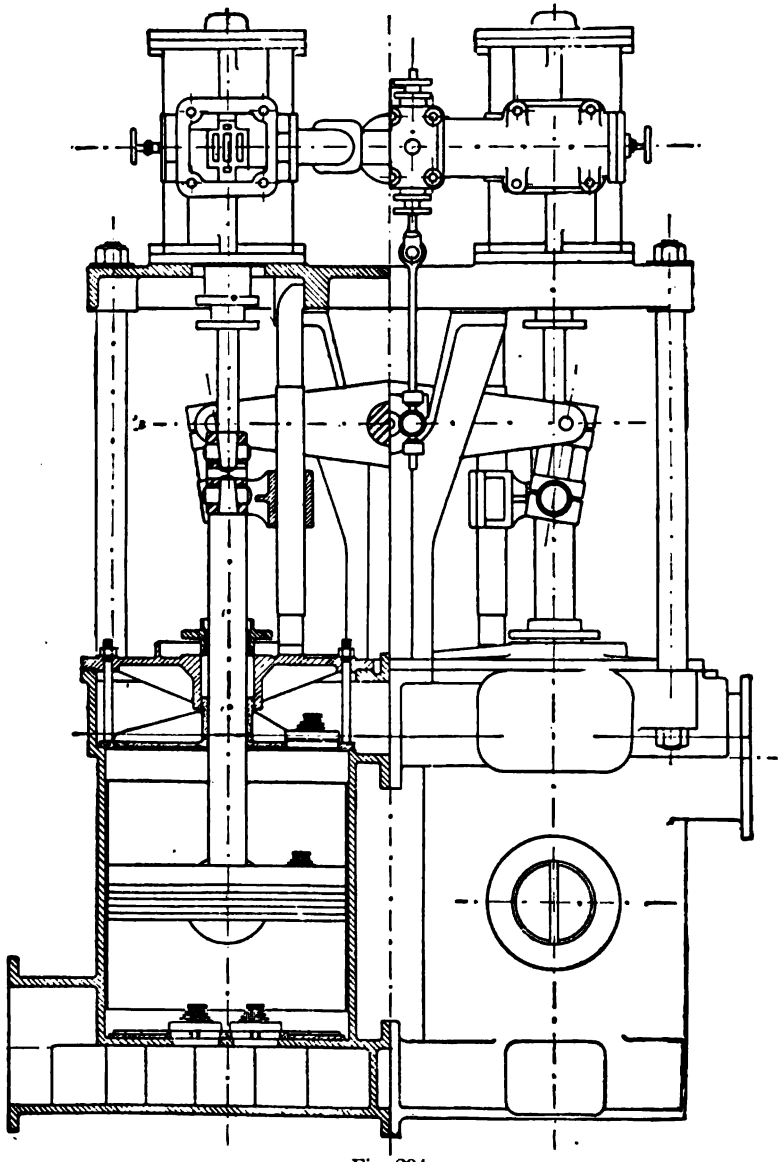


Fig. 294.

§ 164. **Pump Valves.**—The suction and delivery valves should be of such dimensions that the mean velocity of the water through the net sectional area of the valve-seat openings is about 8 to 10 feet per second. The lift of the valve should not, if possible, be more than $\frac{5}{16}$ to $\frac{1}{2}$ inch, and the mean radial velocity of discharge at the circumference of the valve not more than 13 to 15 feet per second. The valves and valve seats are of the same type as those used in air pumps.

To avoid water-hammer, snifting valves are placed between the suction and delivery valves, and an air vessel, the capacity of which is equal to the volume of the air pump, is fitted on the discharge side of the delivery valve.

§ 165. **Suction and Delivery Pipes.**—The mean velocity of the water in the suction and delivery pipes of the circulating pump should be about 8 to 10 feet per second, and should be less in relatively long and small pipes than in short large ones. In order that the circulating pump may be used, in case of emergency, for pumping water out of the ship, the suction pipe from the sea is provided with a branch leading to the bilge, and in German warships to the main bilge suction pipe. This connection must be provided with a non-return screw-down valve. The diameter of this auxiliary bilge suction pipe is about 0·6 to 0·8 that of the main suction pipe.

§ 166. **The Pump Body** of the circulating pump is made of cast iron, of about the same thickness as that of the air pump, and is provided with a gunmetal liner, the thickness of which is about—

$$s = 0{\cdot}015D + {\cdot}25 \text{ inch.}$$

D being the diameter of the pump barrel in inches.

The circulating pump generally forces the water direct through the condenser, and is placed close to the air pump, so that both pumps may be driven off the same beam. Sometimes, however, the air and circulating pumps are driven from different crossheads of the main engine.

§ 167. **Plunger and Pump Rod.**—In double-acting circulating pumps the plunger is made solid, and either packed, like that of the air pump, with hemp, or provided with grooves. In the latter case the depth of the plunger is from two to three times that of the air-pump plunger. The pump rod is made similar to and of the same size as that in the air pump. It is good practice to make the pump rods of the air and circulating pumps of the same size, because then only one spare pump rod need be carried for both pumps.

Centrifugal Circulating Pumps.

§ 168. **General Remarks.**—Of late years centrifugal pumps have been much used to pump the cooling water for the condensers of large engines on merchant vessels. In warships they are employed almost exclusively. They have the advantage of working without shock, and of delivering a uniform supply of water. With these pumps dangerous strains upon the castings, pipes, &c., such as sometimes occur with reciprocating pumps, are entirely avoided.

§ 169. **Suction and Delivery Pipes.**—The velocity v of the water through these pipes is generally taken at 500 to 600 feet per minute. In relatively short, large pipes, a higher velocity of the water may be taken than in long small pipes.

In warships a higher velocity of water through the pipes is often allowed, at maximum power, than in merchant vessels, because it is only in exceptional cases that the maximum power in the former is developed, and weight of both water and material must be economised. The requisite sectional area f in square inches per I.H.P. is given in Table No. 28.

TABLE No. 28.

Sectional Area of Pipes.

Velocity of the Water in feet per minute.	Quantity of Water per I.H.P. per hour in pounds.		
	450	550	650
700	·034 sq. in.	·044 sq. in.	·051 sq. in.
600	·029 „	·035 „	·044 „
500	·024 „	·031 „	·037 „

The total sectional area (F) is obtained from the formula—

$$F = f \times \text{I.H.P.}$$

Both suction and delivery pipes should be made as short, and with as few bends, as possible. If there are two circulating pumps to one main engine, each is generally provided with a separate valve to the sea, and a separate sluice valve on the discharge pipe to the condenser, so that, should one of the centrifugal pumps be damaged, it may be shut off; or blank flanges inserted in the pipes will answer the same purpose. In twin-screw steamers each engine and condenser generally has its separate centrifugal circulating pump. The delivery pipes are often connected together, and fitted with a sluice valve. In case of need one pump may be made to supply water to both condensers.

As the centrifugal pump is also intended, should the necessity arise, to pump out the bilges, the suction pipe is made with a branch leading either to the engine-room bilge, or, as is customary in warships, to the main drain or emergency bilge suction, through a valve which can be either a sluice or a non-return foot valve. The inside diameter of this connection is about 0·6 to 0·8 that of the delivery pipe of the pump. When pumping from the bilges, the pump only gives about half the usual discharge.

§ 170. **Pump Vanes.**—The inner diameter d (see Fig. 295) of the wheel is generally—

$d = 1{\cdot}1$ to $1{\cdot}4d_1$ (d_1 being the diameter of the suction and delivery pipes).

The overall diameter $D = 2$ to $2{\cdot}6d$.

It should be noted that the higher values of D must be chosen, if the head of water is considerable. The overall diameter of the vane wheel is determined with respect to the number of revolutions per minute at which it is designed to run, and to the shape of the vanes. (See Fig. 296.) The clear width B, Fig. 295, at the inner circumference of the wheel is so calculated that the radial speed at this point is about 3·3 to 5·0 feet per second. It varies generally from $B = 0{\cdot}23$ to $0{\cdot}4d$. The clear width b at the periphery of the wheel is either made equal to B, or reduced in proportion to $b = B\frac{d}{D}$.

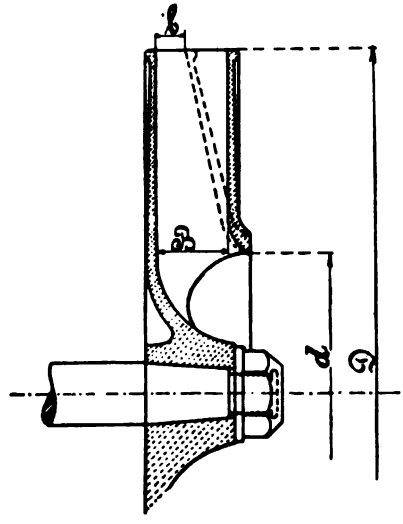


Fig. 295.

The water may enter the wheel either from one or from both sides. (See Figs. 295 and 297.) If it enters at one side only, the wheel exerts an axial thrust which must be taken up by carefully adjusted collars, or some similar arrangement.

The vane wheel is generally of gunmetal, the vanes being cast with it in one piece. Sometimes the vanes are made of sheet copper, and side discs are riveted and soft soldered on to them. These side discs should fit very closely, at least at their inner circumference, to the pump casing, with only about $\frac{1}{32}$ to $\frac{1}{16}$ inch clearance at each side, so that as

little water as possible flows back from the delivery to the suction chamber. The discs are often made to fit into the pump casing with very little clearance at the side of their outer, as well as of their inner circumference.

The shape of the vanes may be varied considerably. The angle α_e (Fig. 296), formed by the vane at the inner circumference of the vane wheel, should be so calculated that the water enters with the least possible shock. Therefore it must be—

$$\tan \alpha_e = \frac{v_r}{v_1}$$

Here v_1 = circumferential speed of the wheel at its inner periphery.
v_r = radial velocity of the water on entering.

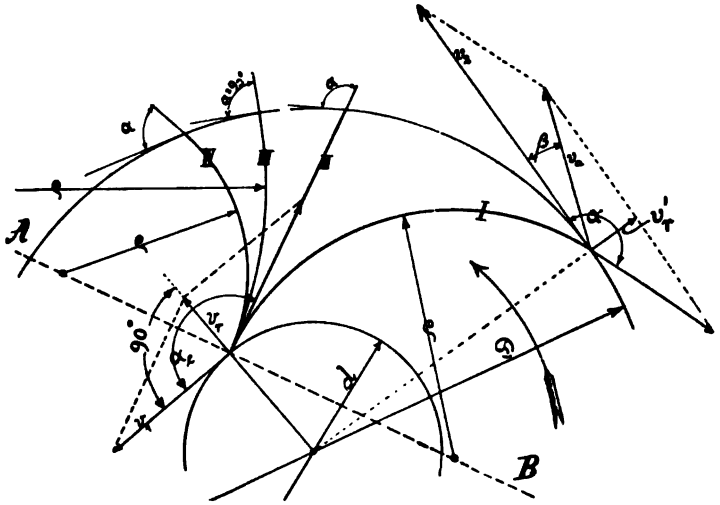


Fig. 296.

To simplify their construction the vanes are generally built up of arcs of circles whose centres lie along the line AB (Fig. 296) at right angle to the blade II.

To obtain a given difference of pressure between the suction and delivery pipes, the curved blade I requires the maximum, and the curved blade IV the minimum circumferential velocity, or number of revolutions per minute. The blades II and III lie in between these values. The circumferential velocity v_2 in feet per second at the outer edge of the wheel is—

$$v_2 = 10 \sqrt{\frac{H}{1 + \frac{\sin(\alpha+\beta)}{\sin(\alpha-\beta)}}}$$

The angles α and β are assumed.

H = total head against which the water is pumped, in feet.

= (actual head of water + head due to friction in the pipe).

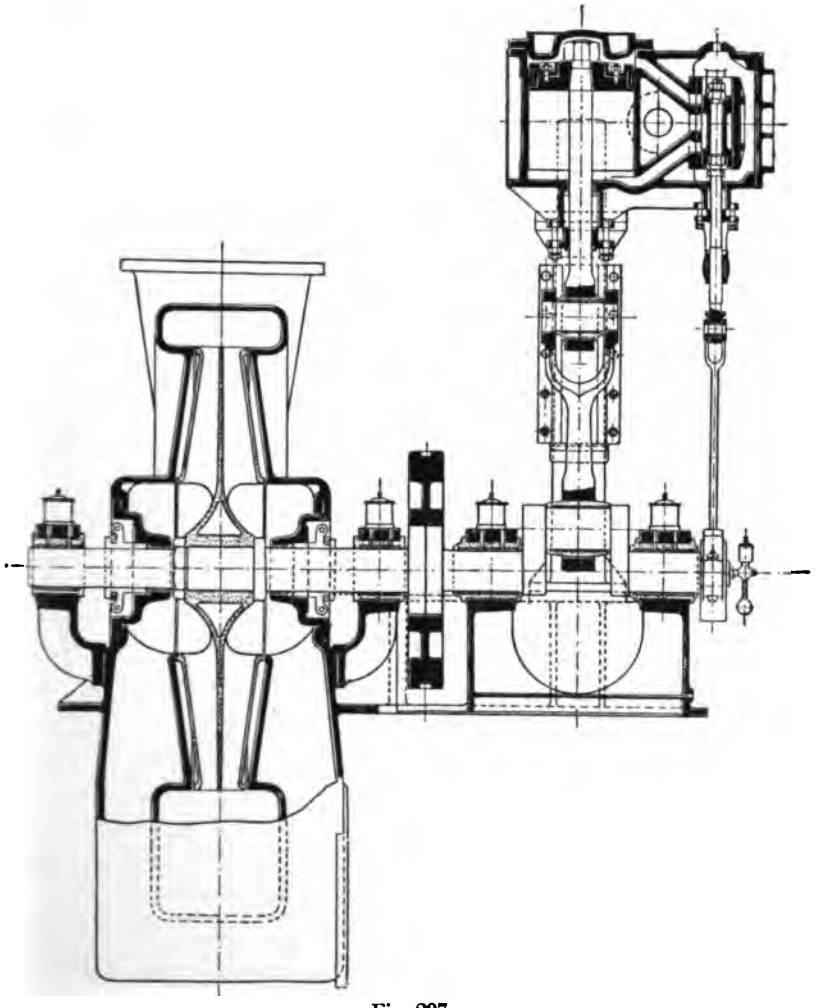


Fig. 297.

The radial velocity v_r (and at the same time the width of the cross section of the wheel at its inner periphery) is assumed; the angle α_ϵ is most easily determined by constructing a velocity parallelogram. The angles α and β must be so chosen that the radial velocity v_r' of the water

leaving the wheel case is about 3·3 to 5·0 feet per second, and that the number of revolutions of the wheel n, when circulating the water through the condenser, is from 140 to 360 per minute. The circumferential

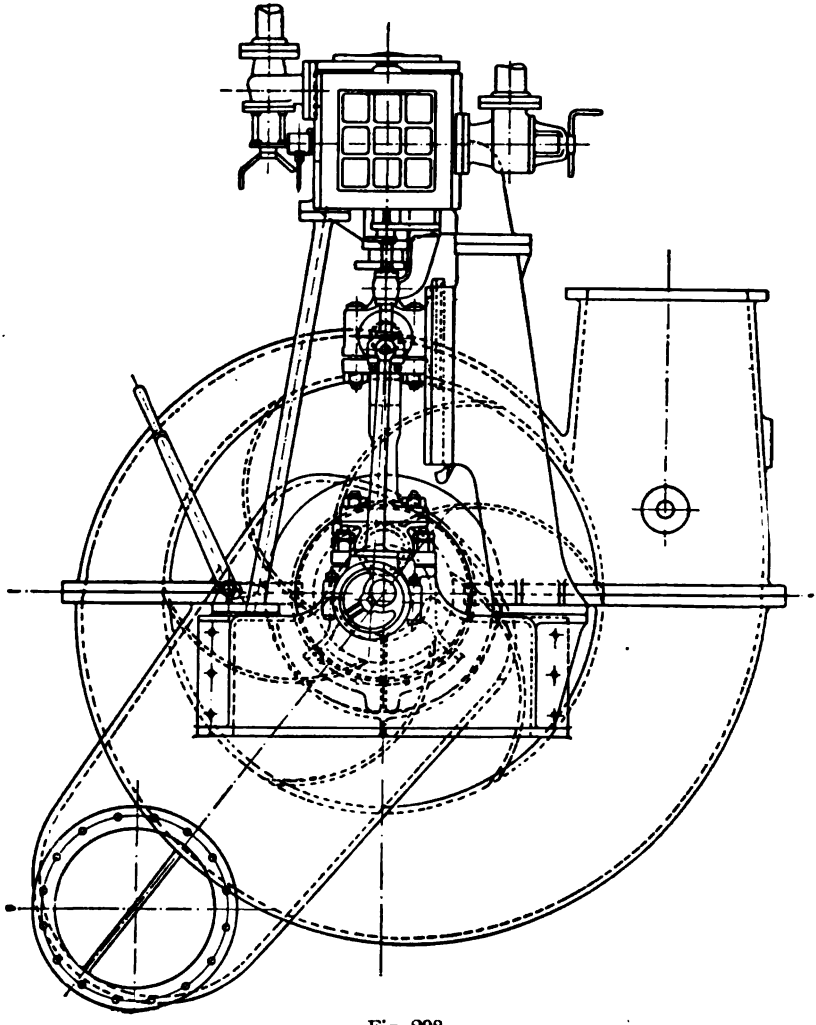


Fig. 298.

velocity v_2 is taken at about 25 to 40 feet per second, and the equivalent head due to friction h_1 under ordinary conditions at about $h_1 = 5$ to 8 feet.

If the circulating pump discharge is below the water line, the

centrifugal pump in ordinary work has only the head due to friction to overcome. In this case $H = h_1 = 5$ to 8 feet. If the discharge is at the height h_2 above the water line, then in ordinary working the head of water will be equal to h_2, and therefore $H = h_1 + h_2$ feet. If h_3 denotes the height of the level of the water outside the ship above the bilge-suction inlet of the circulating pump, then when pumping from the bilges—

$H_1 = h_1 + h_3$ if the discharge is below the water line.
$H_1 = h_1 + h_2 + h_3$ if the discharge is above the water line.

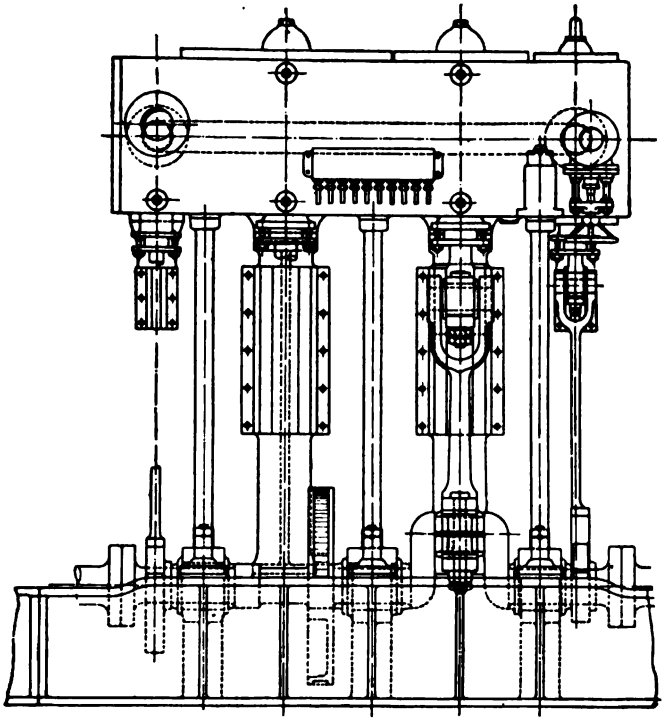


Fig. 299.

§ 171. **The Centrifugal Pump Spindle** is made either wholly of phosphor bronze, or of steel cased in gunmetal, to resist the action of the sea water. The wheel is usually fitted on to the spindle by means of a cone, key, and nut; but it is frequently keyed on to a parallel shaft, and held by a nut against a collar at the other side. (See Fig. 297.) If the water enters at one side only, the wheel is generally overhung with one side bearing in the casing. But if, as is usual in

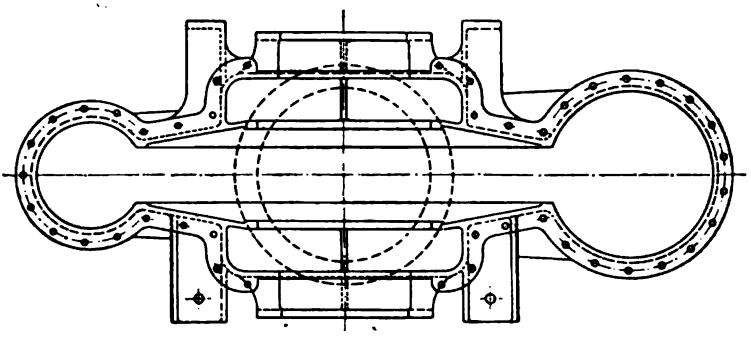


Fig. 300.

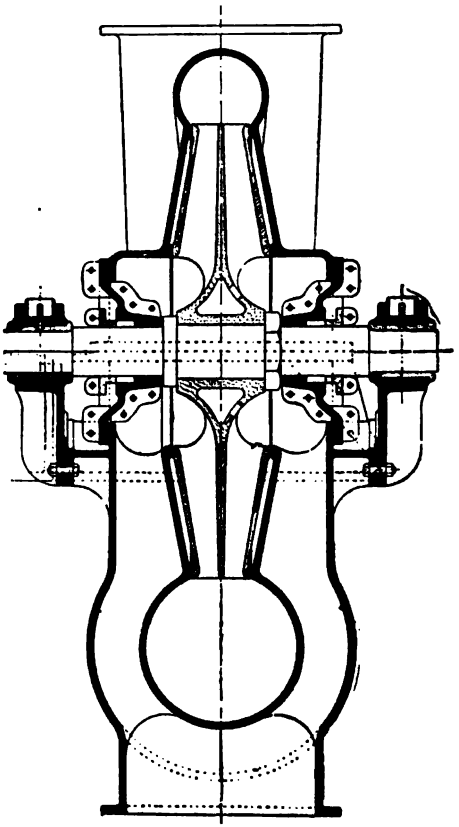


Fig. 301.

larger centrifugal pumps, the water enters on both sides, the spindle is supported by bearings at either side, with stuffing boxes where it passes

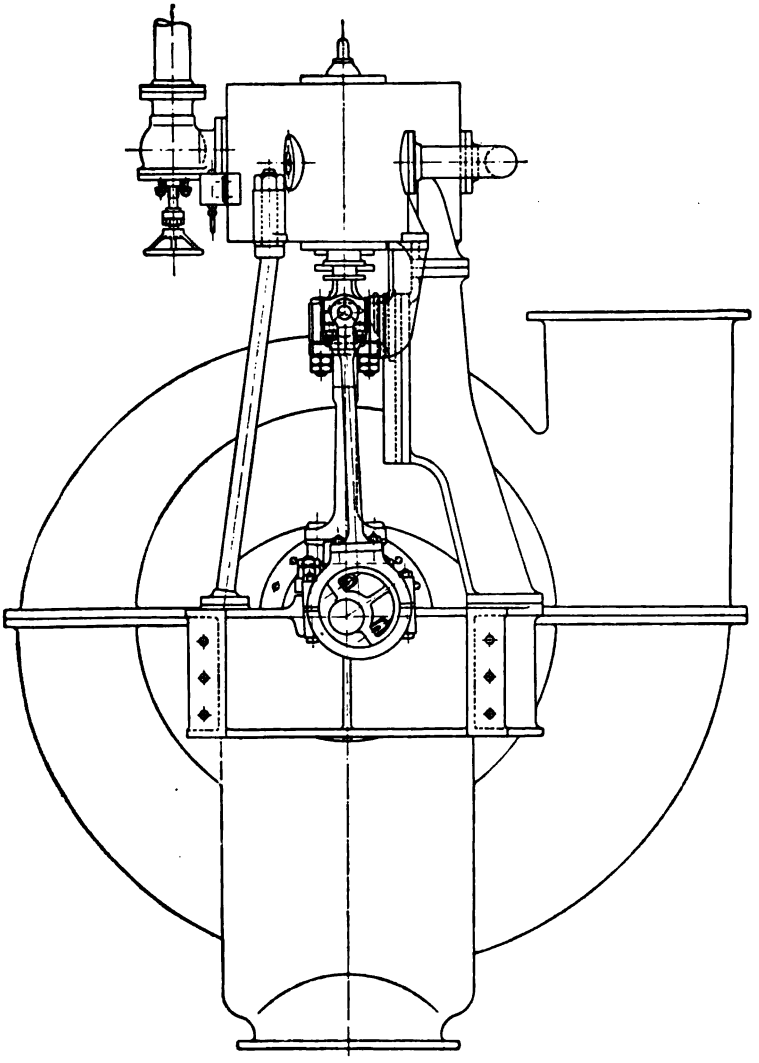


Fig. 302.

through the casing. (See Figs. 297 and 301.) The pump spindle is generally made separate, and bolted by means of a flange coupling to the crank shaft of the engine driving it.

§ 172. **The Pump Casing** of the centrifugal pump is made of cast iron in merchant vessels, and of brass in warships (or occasionally but not often, partly of brass and partly of copper sheeting). In larger pumps it is divided in half across the spindle (Fig. 298), while in smaller pumps, where the water enters at one side only, it has a circular side cover, by the removal of which the wheel can be lifted in and out. The shell of the casing is so shaped that the sectional area of the delivery space is gradually enlarged, until the full area of the delivery pipe is reached. The thickness of the cast-iron casing and of the so-called "vortex chamber" is from $\frac{1}{2}$ to $\frac{3}{4}$ inch, according to the size of the pump. If made of brass, the thickness is about $\frac{3}{16}$ to $\frac{5}{16}$ inch.

§ 173. **Engines for Driving Centrifugal Pumps.**—To determine the dimensions of these engines, the work done by the pump when pumping from the bilges is generally taken as a basis for calculation, because the power required under these circumstances is much greater than that needed for the ordinary work of the pump.

The indicated horse-power of these engines is approximately—

$$\text{I.H.P.} = \frac{Q \times H}{33{,}000} \times 2{\cdot}75,$$

Q being the quantity of water pumped per minute in pounds, and H the total head of water in feet when drawing from the bilge. The above work should be obtained with about 0·75 of the normal boiler pressure. The required number of revolutions n_1 of the centrifugal pump, when pumping from the bilge, is determined from the circumferential velocity necessary to overcome the total head of water H_1, including all frictional losses.

For small pumps these engines are generally made single cylinder, and for large pumps, compound. In the latter case there are frequently two pumps. The principal dimensions are—

For single-cylinder engines $f \times s = \frac{39{,}760 \times \text{I.H.P.}}{p \times n}$ to $\frac{51{,}120 \times \text{I.H.P.}}{p \times n}$*

For compound engines $f \times s = \frac{102{,}240 \times \text{I.H.P.}}{p \times n}$

Ratio of the cylinders is about—H.P. to L.P. = 1 : 3·5 to 1 : 3.

* Circulating pumps of still larger dimensions are not infrequently met with. In these the steam supply has to be considerably throttled, and consequently the pumps are much larger than is necessary, and their steam consumption is correspondingly high.

Here f = the area of the piston in square inches (in compound engines the area of the low-pressure cylinder).

s = stroke of the piston in feet.

p = absolute boiler pressure in pounds per square inch.

n = number of revolutions per minute when pumping from the bilges (*i.e.*, against max. head).

I.H.P. = the indicated horse-power of the engine when pumping from the bilges (*i.e.*, against max. head).

§ 174. **Particulars of Surface Condensers.**—The following table gives particulars of surface condensers taken from actual engines :—

TABLE No. 29. *Particulars of*

Type of Ship.		Small Cargo Boat.	Medium-sized Cargo Boat.	Large Cargo Boat.	Mail Steamer.
Number and I.H.P. of main engines		1 × 700	1 × 2000	2 × 2100	2 × 4500
Number of revolutions per minute		75	70	85	90
Number of condensers and Cooling surface (in sq. ft.) for *each* main engine		sq. ft. 1 × 1076	sq. ft. 1 × 3012	sq. ft. 1 × 3120	sq. ft. 1 × 6025
Centrifugal circulating pumps.	Number of vane wheels per main engine	—	1	1	1
	External diameter of each wheel	—	30 in.	43·3 in.	51 in.
	Internal do. do.	—	2 × 13·7 in.	2 × 15 in.	2 × 19·6 in.
	Number of circulating engines per main engine	—	1	1	1
	Cylinder diameter of circulating engines	—	8·6 in.	9·8 in.	—
	Stroke of circulating engine	—	7 in.	7 in.	—
Reciprocating circulating pumps.	Number per engine	1	—	—	—
	Diameter of cylinder and stroke	Diam. 9·8 in. Stroke 15·7 in.	—	—	—
Air pump.	Type and number per main engine	1 attached to main engines.	1 attached to main engines.	1 attached to main engines.	1 Blake
	Diameter of steam cylinder	—	—	—	2 × 12 in.
	Diameter of pump cylinder	15·7 in.	26·7 in.	23·6 in.	2 × 25 in.
	Stroke	15·7 in.	23·6 in.	19·6 in.	18 in.
	Volume swept through per I.H.P. per minute cubic feet	190	268·3	202	$n = 17$ 39
	Ratio $\frac{\text{I.P. cylinder}}{\text{Air-pump cylinder}}$	21·1	15·5	18·5	33·8 (for each pump cyl.)

Surface Condensers.

"Deutschland."	"Kaiser Wilhelm II."	Russian Cruiser "Bogatyr."	Building for the German Navy in 1903.			
			Gunboat.	Small Cruiser.	Large Cruiser.	Battleship.
2 × 16500 75	4 × 10000 80	2 × 10000 150	2 × 650 180	2 × 5000 150	3 × 6330 120	3 × 5330 115
sq. ft. 1 × 21520	sq. ft. 1 × 11700	sq. ft. 1 × 10760	sq. ft. 2 at 807	sq. ft. 2 × 5380	sq. ft. 3 × 7530	sq. ft. 3 × 6260
2	1	2	1	1	1	1
47 in. 2 × 19·6 in.	51 in. 2 × 19·6 in.	47 in. 2 × 23·6 in.	19·6 in. 2 × 7·5 in.	39·3 in. 2 × 19·6 in.	43·3 in. 2 × 19·6 in.	47·2 in. 2 × 23·6 in.
1	1	1	1	1	1	1
11 × 19·6 in.	12 in.	2 × 7·8 in.	4·3 × 7·8 in.	7·5 × 11·8"	7 × 12 in.	6·3 × 11 in.
11·8 in.	9·8 in.	9·8 in.	4·3 in.	7 in.	7·4 in.	9·8 in.
—	—	—	—	—	—	—
—	—	—	—	—	—	—
1 Blake	1 Weir	1 Weir	2 attached to main engines	2 attached to main engines	Duplex	1 Weir-Duplex
2 × 18 in.	2 × 10 in.	2 × 12·5 in.	—	—	2 × 12 in.	2 × 10 in.
1 × 44 in. 24 in.	2 × 33 in. 31 in.	2 × 37 in. 15 in.	12 in. 7·8 in.	18·5 in. 15 in.	2 × 31·5 in. 19·6 in.	2 × 31 in. 14 in.
$n = 15$ 39 35·3 (for each pump cyl. referred to both L.P. cyls.)	$n = 15$ 31·4 39·1 (for each pump cyl.)	$n = 15$ 28 17·0 (for each pump cyl. referred to both L.P. cyls.)	295 both pumps 9·8 (for both pump cyls. together)	139·4 both pumps 20·6 (for both pump cyls. together)	$n = 15$ 42 16·5 (for each pump cyl.)	$n = 20$ 46 22·8 (for each pump cyl.)

Feed Pumps.

§ 175. **Classification.**—Boilers are fed by pumps driven from the main engine and by steam pumps, and in small vessels even injectors or hand pumps are used. It is now made compulsory to have at least two independent feed systems for each set of boilers, each capable of providing the full amount of feed water required to work the boilers. In practice, however, there are often three feed-supply systems entirely independent of one another, and in large ocean-going steamers even more, so as to render impossible, as far as may be, any breakdown in the feed arrangements. Where steam is required for electric lighting, winches, &c., when the main engines are not running, two independent feed systems must be provided. In such cases, two independent feed pumps are required; or for small and medium-sized boilers, one pump and one injector are sufficient.

§ 176. **Amount of Feed Water required.**—Practice has shown that the number of cubic feet of feed water per I.H.P. per hour q, which must be supplied to the boiler, may be taken at—

$q = 0{\cdot}32$ cubic feet or 20 lb. for compound engines.

$q = 0{\cdot}24$ to $0{\cdot}26$ cubic feet or 15 to 16·5 lb. for triple and quadruple expansion engines.

These quantities include not only the steam required for the main engine, but also that for the auxiliary engines and steam heating, as well as losses due to radiation, &c.

1. Pumps Driven Direct from the Main Engine.

§ 177. **General Arrangement.**—Engines up to 100 I.H.P. have frequently only a single feed pump. Larger engines have generally two feed pumps of equal size, each capable of supplying the full amount of feed water required for the boilers, when the ship is going at full speed. The pumps are generally placed close to the air and circulating pumps, and are driven from the crosshead in common with the other pumps. In engines running at more than 200 revolutions per minute, the feed pumps are often driven by a worm and worm wheel from the crank shaft. They are then placed horizontal or inclined, and parallel to, or on the longitudinal axis of the engine. If

there are two feed pumps, each suction and delivery is as a rule fitted with a valve, so that either pump may be worked independently. To prevent abnormal strains occurring, due to excessive rise of pressure in the pumps, relief valves are fitted, either on the pump itself, or on the delivery pipe between the pump and delivery valve. The spring of the relief valve should be set so as to allow the valve to lift at about 1·3 times the boiler pressure.

§ 178. **Size of Feed Pumps.**—Each separate pump must be of such dimensions that it can easily deliver the full quantity of feed water required for the boiler. Experience has shown that the pumps are large enough if they are of such dimensions that, with an efficiency = unity, each pump can deliver from one-and-a-half to twice the amount of feed water required.

Pumps driven from the main engine have plunger pistons, and hence are single-acting. The volume of each pump works out at—

$$f \times s = 10{\cdot}5\frac{\text{I.H.P.} \times q}{60n} \text{ to } 14\frac{\text{I.H.P.} \times q}{60n}$$

where f = area of the plunger in square inches, s = stroke in inches, q = quantity of feed water per I.H.P. per hour in pounds.

In twin-screw steamers the pumps for each engine are designed to deal with from 1·3 to 1·5 times the amount of feed water required for each engine.

§ 179. **The Barrels and Valve Boxes** of the feed pumps are generally of cast iron in cargo boats, and always of gunmetal in warships. As a rule the barrel and valve boxes are cast separately, and joined by flanges, in order that, if one part is damaged, it alone has to be replaced. If the internal diameter of the pump is larger than that of the plunger, the valve box must be connected to the highest part of the clearance space thus left, in order that no air cushion may be formed. The thickness of the barrel δ is taken at $\delta = \frac{d_1}{10} + {\cdot}28$ inch for cast iron, $\delta = \frac{d_1}{12} + {\cdot}16$ inch for gunmetal, d_1 being the internal diameter of the pump cylinder. Pump rods in cargo boats are made of steel, and frequently cased with gunmetal, the cases being held in position by cap nuts. The pump rod is made tight with an ordinary stuffing box packed with hemp or metallic packing. Each feed pump driven from the engine has an air vessel, the capacity of which is about 2·5 times the volume swept through by the plunger.

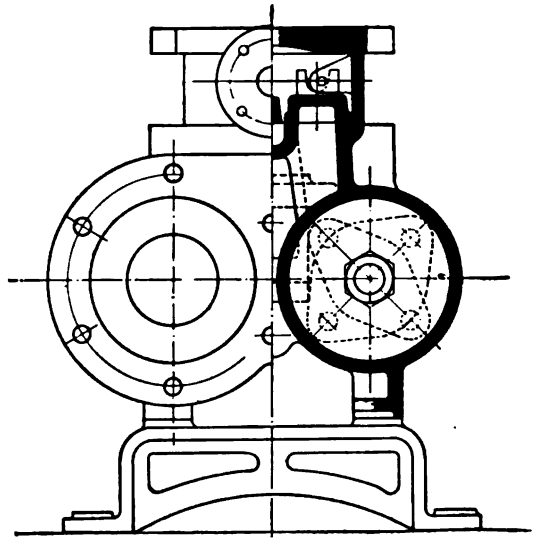


Fig. 303.

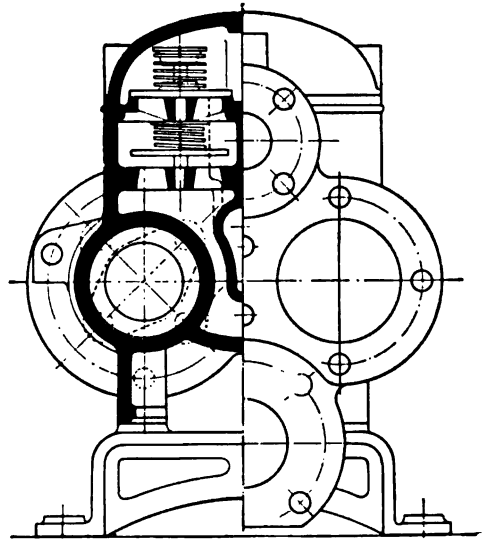


Fig. 304.

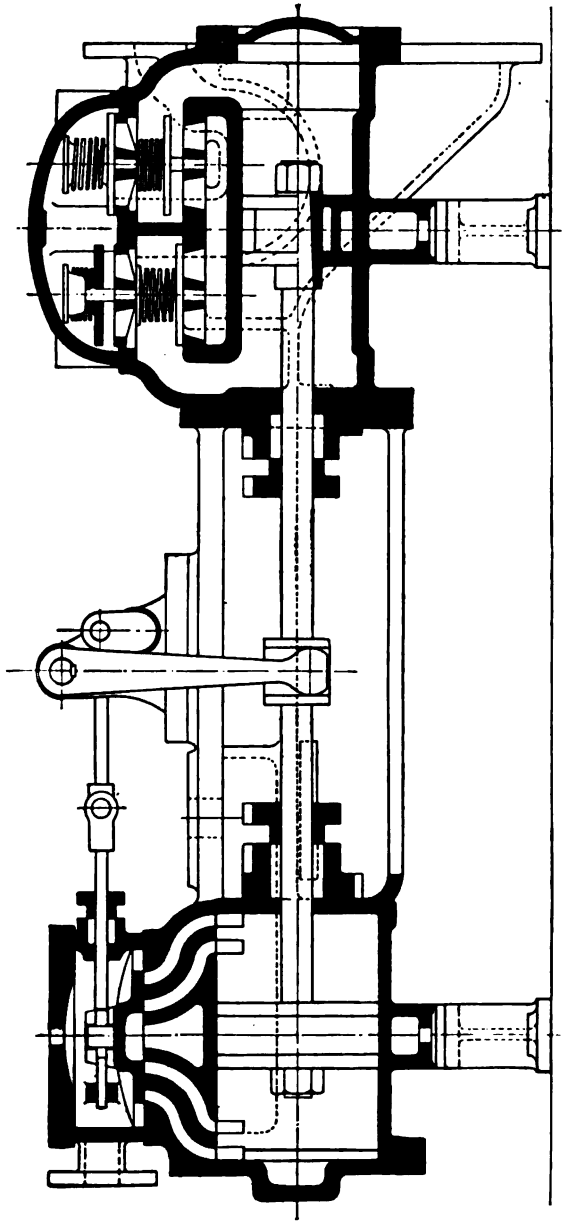


Fig. 305.

§ 180. **The Pump Valves**, in smaller pumps up to about $2\frac{1}{2}$ inches diameter, are usually of the mushroom or ball valve type; in larger pumps they are of the Kinghorn or of a similar type. If mushroom valves are used for larger pumps, there should be several small instead of one large valve, the object being to have sufficient sectional area of valve opening, while reducing the lift (about $\frac{1}{8}$ to $\frac{5}{32}$ inch) and weight of valve to a minimum. To make the valves close quickly at the change of stroke, they are sometimes fitted on the back with helical springs of "delta" metal. The valve seats are made separately, and fitted into the valve chests, so that they may be easily taken out and renewed. If solid valves are fitted the valve seatings are made about $\frac{1}{8}$ to $\frac{5}{32}$ inch wide. Between the suction and delivery valves a small snifting valve is usually fitted, and a small air cock is sometimes placed in the lower part of the air vessel, the cock being connected to the hot well. This not only allows the air collected there to escape, but at the same time prevents any water being wasted from a too free use of this cock.

§ 181. **Velocity of the Water.**—The net sectional area of the openings in the valve seat is calculated to allow a mean velocity of the water through it of about 6·5 to 8·25 feet per second. The radial velocity of the discharge at the circumference of the valve should be about 20 to 26 feet per second, and the lift given to the valve $\frac{1}{8}$ to $\frac{5}{32}$ inch, to prevent shock or jarring at the change of stroke.

Suction and Delivery Pipes.—The diameters of these are so calculated that the mean velocity of the water in the suction pipe is about 7 feet per second, and in the delivery pipe about 10 feet per second.

Remarks.—In calculating the different areas such as the bore of the pipes and the net openings of the valves, the mean speed and sectional area of the pump plunger must always be taken as a basis, and it is unnecessary to consider whether the pump is single or double acting.

2. Independent Feed Pumps.

§ 182. **Steam Pumps** are generally used, in small engines, as auxiliary feed pumps only, while in larger engines they are frequently employed to pump the whole of the feed water required. In the latter case the working of the pump may be controlled automatically, and its speed made to correspond to the quantity of feed water required at any given time. The type of steam pump most generally used is either the Duplex or Simplex. These pumps, which may be either vertical or horizontal, are double-acting, and have no rotating parts. The steam slide valve is driven either directly or indirectly by the piston rod. The diameter of the steam cylinder is about 1·4 to 1·7 that of the corresponding pump cylinder. For the

steam cylinder cast iron is used, and for the pump cylinder either cast iron with a gunmetal liner and valves, or the pump and valve box are both made of brass, and cast in one piece. It is best to have the pump plunger and pump rod of phosphor bronze, or some similar material.

§ 183. **Duplex Pumps.**—Figs. 303 to 305 show a horizontal, and Figs. 306 and 307 a vertical Duplex pump, as made by the Worthington Pump Co. There are two steam cylinders of equal size, each one driving a pump plunger; the movement of the pistons corresponds to that in an engine with two cranks at 90°, and two eccentrics at an angle of 90° with the cranks. Each piston rod works the slide valve of the other steam cylinder. One of the levers transmitting the valve motion must be single-armed, the other double-armed, as can easily be demonstrated if imaginary cranks and eccentrics are drawn for both gears. The valve of each cylinder is an ordinary D slide or piston valve, but the cylinder has five ports. The two outermost lead to the two ends of the cylinder, and are for admission only; the two inner open into the bore of the cylinder, and take the exhaust; the central port is also for the exhaust. Outside and inside lap, $\frac{1}{32}$ to $\frac{3}{16}$ inch; utmost travel of the valve = 2 × (lap + height or breadth of a port); mean speed of the steam through the ports, 4,000 to 6,000 feet per minute. Maximum number of double strokes per minute = 40 to 50, but more generally 20 to 30. The motion of the valves is often regulated by adjustable nuts on the valve rods, so that the steam ports may the longer remain fully open, and the stroke of the pump be varied.

As the motion of the valves is alternating, it follows that both cylinders are interdependent, and must work at the same time. To prevent the steam piston from striking against the cylinder covers, the exhaust ports opening into the bore of the cylinder are closed by the piston before it reaches the end of its stroke, and the steam remaining in the clearance space thus forms a cushion. In the new Duplex pumps made by the Blake & Knowles Steam Pump Co., the openings of the outer ports in the valve face of the steam chest are not outside, but at the side of the openings of the inner ports. The former serve for admission only, the latter for admission and exhaust. The length of the valve is thus diminished, but at the cost of the breadth. There is also an arrangement for regulating the compression by means of four small compensating valves between the outer and inner ports. A somewhat large Duplex pump for a duty of 120 tons per hour, which may be used as a donkey pump or for the ash-ejector, is shown at Figs. 308, 309, Plate XIV. It has a balanced slide valve, and a pump plunger with a packing ring of white metal. Each of the steam cylinders has a diameter of 14 inches, each of the pump cylinders of 9 inches. Stroke 12 inches. In

the newer kinds of pumps, which work with clean cold water, leather rings are often used for packing the pump plungers.

§ 184. **Simplex Pumps.**—These have only one steam and one pump cylinder. Their gearing and method of working are so complicated that they can hardly be explained without a model. Two such pumps, each independent of the other, are frequently combined into a twin Simplex pump. (See Fig. 310.)

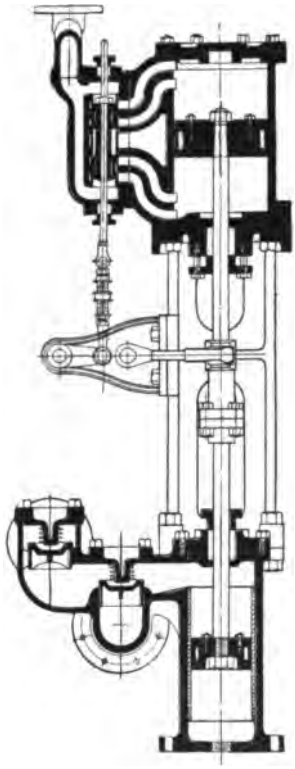


Fig. 306.

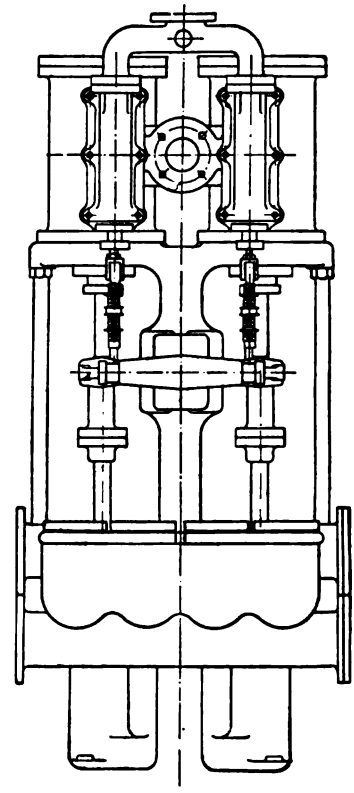


Fig. 307.

All Simplex pumps are worked by what may be known as *indirect valve gears*. A small auxiliary steam valve, driven from the piston or piston rod, works the main distributing valve controlling the steam supply to the cylinder. If there is only one direct driven valve, dead points cannot be avoided, further it is impossible to start the pump in any position, and when working it is very liable to come suddenly and unexpectedly to a standstill.

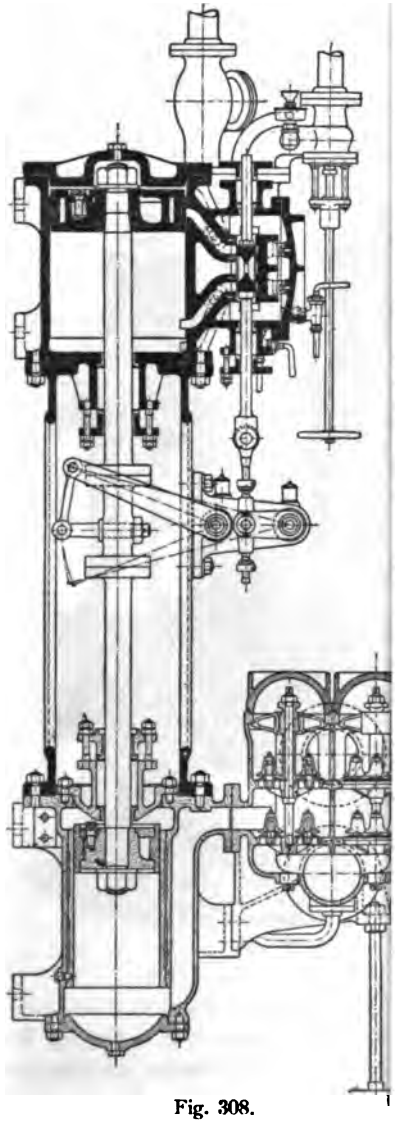


Fig. 308.

§ 185. **Weir Pumps.**—(See Fig. 310, showing a twin Simplex pump.) For steam distributing valves of Weir pumps see Fig. 311. The semicircular main slide valve d has rounded ends m_1 and m_2 at either side, which work to and fro in the horizontal auxiliary steam cylinders h_1 and h_2 of the valve chest. In the vertical recess at the back of the main valve, an auxiliary slide valve c, driven from the piston rod through a single-armed lever, works parallel to the cylinder axis. Of the two steam passages, a_1 leads to the bottom end, a_2 to the

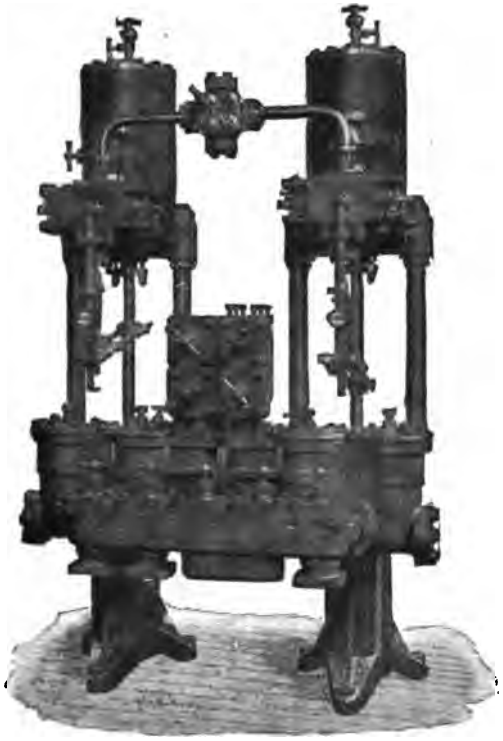


Fig. 310.

top end, of the steam cylinder. In the position shown at Fig. 311 the lower part of the valve is receiving live steam, while the upper part communicates with the exhaust b, in this position therefore the piston ascends. It is only after it has returned through about 0·75 of the stroke, that it takes the auxiliary valve c with it. The latter gradually shuts off e_1, so that the steam in the bottom end of the steam cylinder expands. At the same time the passage f_1 leading to the auxiliary

cylinder h_1 is opened to exhaust by the auxiliary valve c, and the passage f_2 leading to the auxiliary cylinder h_2, is opened to admit live steam, so that as the pressure in h_2 is above that of the atmosphere, the main valve d is forced to the other side of the valve chest. In this way, a_2 being open to live steam and a_1 open to exhaust, steam is admitted to the piston at the top end, and drives it down. To prevent the main valve d striking the cover of the auxiliary steam cylinder h in its travel, it is so arranged that it closes the port f_1 before it has reached its extreme position, and the steam in h^1 is compressed, forming a steam cushion.

Weir pumps, similar to Blake pumps, can be regulated while running, by adjusting the lock-nuts on the valve rod. The steam and pump cylinders are braced together by strong steel columns. The pump cylinders or barrels are connected by suction or delivery valve boxes, or by means of both.

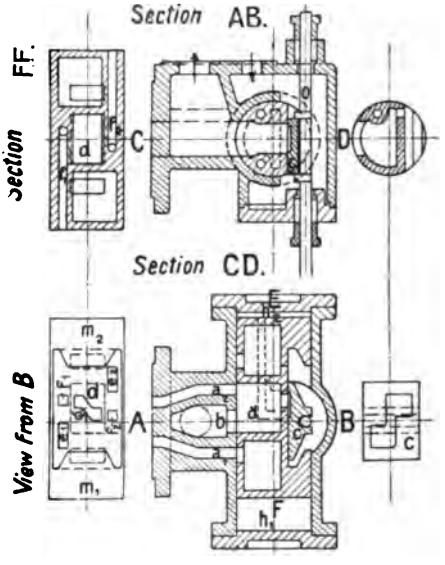


Fig. 311.

In Weir's twin air pumps (see Fig. 293) the two steam cylinders are worked by one main valve, common to both. The photograph shows clearly the action of the lever working the corresponding auxiliary valve; and also the valve box in front, with the two jointed covers at the side, and the two expansion valves.

§ 186. **Blake Pumps.**—For steam distributing valves see Fig. 313. In these pumps the steam is distributed through a slide valve d, having a hollow circular back, upon which works the auxiliary valve h, which

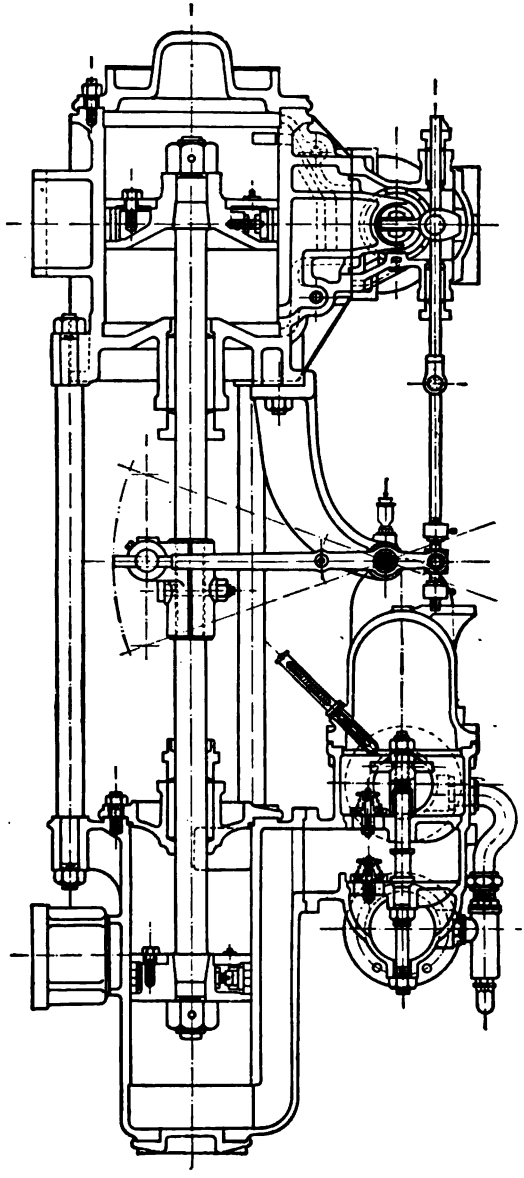


Fig. 312.

can be rotated. The ends are shaped like pistons, and work in the horizontal passages *m m* of the valve chest. Towards the end of the stroke of the steam piston, the valve rod *o* turns the auxiliary valve *h* once, through lever *n*; and thus the recesses *e e* in the auxiliary valve *h* form a connection on one side of the valve between the admission steam passage *i*, and on the other side between the exhaust passage *l* and the auxiliary steam cylinders *m m*. The auxiliary valve *h* is thus forced to the other side, and takes with it the main valve *d*. Towards the end of the stroke the main steam ports *a* are covered by the piston rings, and the steam remaining in the cylinder is compressed as in the Duplex pump. At the beginning of a stroke the steam is admitted through the

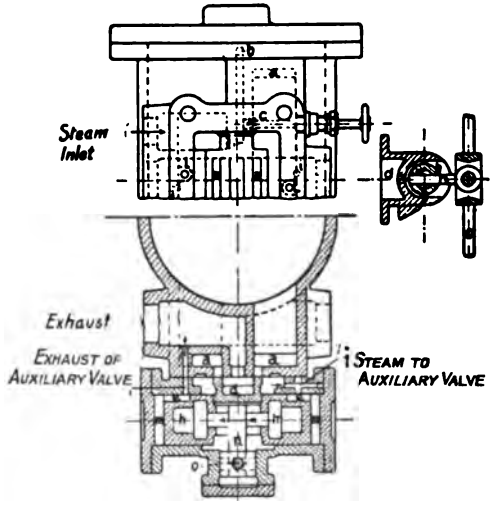


Fig. 313.

auxiliary ports *b*, which are connected to the main steam ports through the compression valve *c*.

To Adjust the Pump.—The pump is set to work dead slow against a pressure, and the lock nuts on the valve rod adjusted so that the steam piston knocks against the top and bottom cylinder covers. These positions are marked on the columns of the pump, and the nuts brought closer together, until the pointer on the piston rod is from 0·2 to 0·4 inch short of the marks.

The steam and pump cylinders are connected by three strong wrought-iron columns. These pumps are set up either singly, or, like Weir pumps, in sets of two side by side, and connected by the suction and delivery valve boxes.

Auxiliary Pumps.

§ 187. **Bilge Pumps Driven by the Main Engine.**—These are as a rule made similar to the feed pumps, and the diameter of the pump plungers is the same. Ships with engines of less than 100 I.H.P. have generally only one bilge pump. Ships with larger engines have at least two, one of which draws water direct from the engine-room bilge,

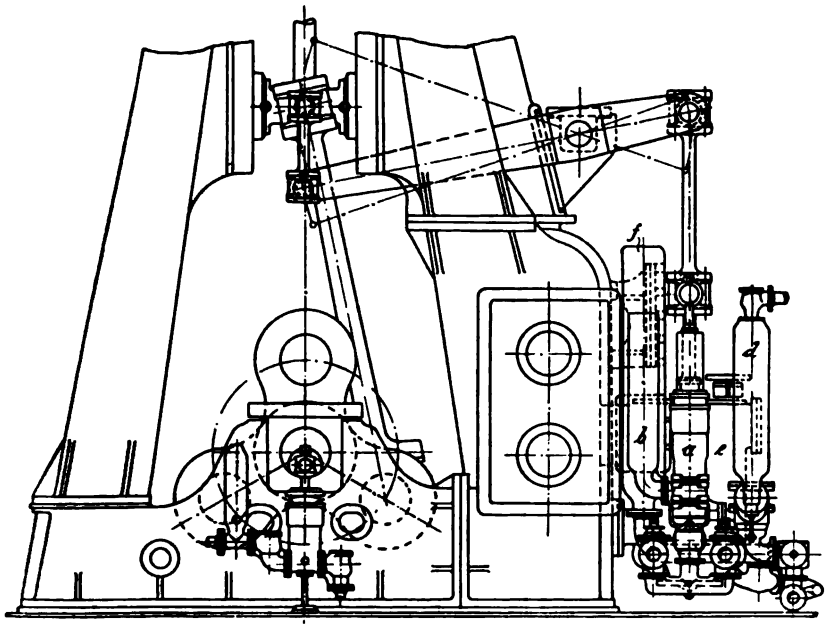
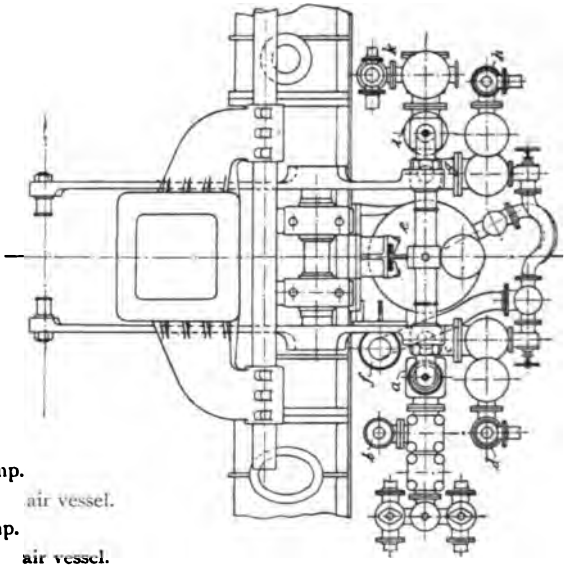


Fig. 314.

and the other is arranged to draw from all the compartments of the ship. The pump barrel and valve boxes are generally of cast iron, but occasionally of brass. The valves are usually rubber flap valves, or in small pumps solid gunmetal mushroom valves. The suction and delivery valves, instead of being placed one above the other, are set side by side, and can be reached through separate covers. The advantage of this arrangement is that the suction valve can be easily examined. The



a. Bilge pump.
b. ,, air vessel.
c. Feed pump.
d. ,, air vessel.
e. Air pump.
f. ,, air vessel.
g. Feed pump.
h. ,, air vessel.
i. Sanitary pump.
k. ,, air vessel.

Fig. 315.

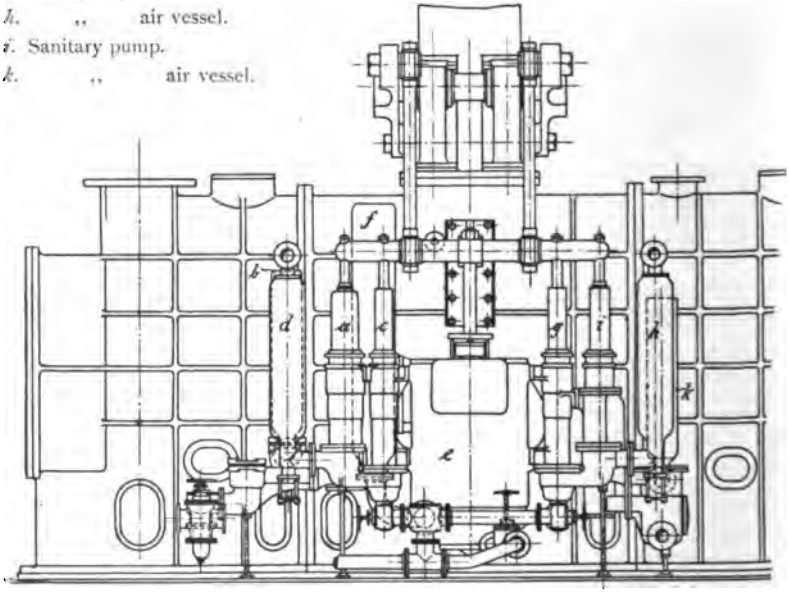


Fig. 316.

gunmetal plunger is hollow, and is fixed to the steel pump rod by means of a coned end, and a cap-nut underneath. The pump rod is connected to the crosshead common to all the pumps by means of a cone and nut, in the same way as the feed pump rod. The air vessel is of the same capacity as that of the feed pump. In large ships the delivery pipe leading from the air vessel to the discharge valve in the side of the ship is often so arranged that it can be shut off at the air vessel by an automatic valve; by this means, when the pump valve box is opened for repairs, the water in the discharge pipe does not run back. This valve must of course open freely when the pump is working.

§ 188. **Sanitary Pumps.**—These are generally of the same dimensions, and arranged in the same way, as the feed and bilge pumps, unless special conditions render necessary a larger quantity of water than usual. In small ships carrying two bilge pumps, one of them is generally arranged to act either as a bilge or as a sanitary pump.

§ 189. **Arrangement of Pumps.**—Figs. 314, 315, 316 show the method of arranging the pumps in an ordinary merchant vessel, when they are driven from the main engine through levers. The two levers drive a crosshead through two connecting links. The air pump rod *e* is connected to the middle of the crosshead; immediately to the right and left of it are the two feed pumps *c* and *g* (with steel plungers), and outside these again the bilge pump *a* on the left, the sanitary pump *i* on the right, both with gunmetal plungers.

§ 190. **Separate Steam-driven Pumps.**—Where these are used the bilge pump, sanitary pump, fire pump, and circulating pump may be either of the Duplex or Simplex type. As a rule, in smaller ships, the donkey feed pump is arranged to perform these various functions. In large ships, on the other hand, special pumps for the bilges, for delivering sea water on deck, &c., are provided, so that the various demands may be satisfied at the same time, and the donkey feed pump not soiled with the bilge water. The latter is generally used to work the ash ejector, and is arranged in the same way as the steam feed pump. The principal difference between them is in the heavier or lighter construction of the various parts, according to the purposes for which either pump is required. To pump out the ballast tanks, if there are any, special steam pumps are generally fitted. These are usually Duplex, but sometimes centrifugal pumps or pulsometers are used. See Lloyd's Rules concerning pumps, pipes, &c., § 193, page 329.

Pump Rods.

§ 191. **General Remarks.**—The pressure on the plunger of the air pump is taken as a basis for determining the dimensions of the rod driving the pump, when worked from the main engine. This pressure, including the friction of the stuffing boxes and plunger, is assumed to be 28·5 lb. per square inch of piston area. It also serves as a basis for designing the circulating pump rods. The allowable stress in the bearing and connecting-rod bolts of the pump-rod gear is taken at 3,500 to 6,500 lb. per square inch. The stress allowed in the rocking beams, which have to withstand a bending strain, can be taken as 4,000 to 5,000 lb. per square inch.

The sizes of the various pins and journals are calculated by allowing for a pressure on their working surfaces of from 425 to 700 lb. per square inch. Their length is generally slightly greater than their diameter.

The crosshead or beam to which the air pump, feed water, and bilge pumps are attached, is chiefly subjected to bending strains, and its sectional area must be determined in accordance with these strains in each particular case. The allowable stress in it, due to bending, may be about 3,000 to 5,000 lb. per square inch.

Feed pumps are generally fitted towards the forward end, and bilge pumps towards the after end of the pump crosshead, but to ensure a more uniform strain upon the rods, one of the feed and one of the bilge pumps are frequently placed side by side.

§ 192. **Different Pump Arrangements taken from actual practice.**—In the following tables the pump systems of different types of ships are classified, to show from whence the pumps of each system draw, and where they deliver to :—

(*a*) Table No. 30.—Pump arrangement of a river tug.

(*b*) Table No. 31.—Pump arrangement of a small cargo boat.

(*c*) Table No. 32.—Pump arrangement of a large cargo and passenger steamer.

(*d*) Table No. 33.—Pump arrangement of a large cruiser of the Imperial German Navy.

The different uses to which the pumps can be put are plotted on what is known as a "pump diagram."

TABLE No. 30.

Steam Tug—Jet Condensing.

Pump.	Draws from	Delivers to
Air pump (one)	Condenser Bilge	The sea.
Engine feed pumps (one or two)	Hot well	Main feed pipe (possibly through a surface feed heater).
One independent steam pump	The sea Hot well Bilge Ballast tanks	To deck. Auxiliary feed pipe. The sea. Ballast tanks.
Injector	The sea	Auxiliary feed pipe.
Engine bilge pump (one or two)	The sea Bilge Ballast tanks	The sea. Ballast tanks.
Bilge ejector	Bilge	The sea.

TABLE No. 31.

Small Cargo Boat—Surface Condensing.

Pump.	Draws from	Delivers to
Circulating pump (double-acting, driven by main engine)	The sea Engine-room bilge	Through the condenser to the sea.
Air pump (driven by main engine, single-acting)	Condenser	Hot well.
Two engine-driven feed pumps	Hot well Condenser	Main feed pipe (possibly through surface feed heater).

TABLE No. 31—*continued.*

Pump.	Draws from	Delivers to
One engine-driven bilge pump	Engine room bilge Bilge piping	The sea.
One engine-driven sanitary pump	The sea Engine room bilge Bilge piping	The deck. Sanitary tank. The sea.
Independent steam pump	The sea Bilge piping Condenser Boiler Reserve feed tanks in double bottom Ballast tanks	Auxiliary feed piping. The deck. Through the condenser to sea.
Ballast pump	The sea Ballast tanks Bilge piping	The sea. Ballast tanks. Through the condenser.
Injector	The sea Reserve feed tanks in double bottom	Auxiliary feed piping.
Hand pump (for filling the boiler)	The sea Reserve feed tanks in double bottom	Auxiliary feed piping.

TABLE No. 32.

Large Cargo and Passenger Steamer—Two Engines, Surface Condensing.

Pump.	Draws from	Delivers to
Circulating pump (one centrifugal pump to each engine)	The sea Engine room bilge	Through the condensers of both engines to the sea. Auxiliary condenser.
Air pump (one steam-driven air pump to each condenser)	Both condensers	Hot well.

TABLE No. 32—*continued.*

Pump.	Draws from	Delivers to
Two steam-driven feed pumps (delivering to feed heater)	Hot well Reserve feed tanks in double bottom Condenser	The sea. Feed heater. Main and auxiliary feed piping.
Two steam-driven main feed pumps	Feed heater Hot well Boiler Condenser Reserve feed tanks in double bottom	The sea. Feed heater. Main and auxiliary feed piping.
Two steam donkey pumps	The sea Boiler Condenser Main bilge piping Auxiliary bilge piping Ballast tank suction Cooling water from refrigerating engine	The sea. Ash ejectors. Deck and fire hose. Sanitary tank.
One ballast pump	The sea Main ballast piping Main and auxiliary bilge piping	The sea. Main condenser. Auxiliary condenser.
Auxiliary condenser circulating pump	The sea	Through auxiliary condenser to the sea.
Engine bilge pumps (two per engine)	Engine-room bilge Main bilge piping	The sea.
Sanitary pumps (one per engine)	The sea	Deck and fire hose. Sanitary tank. Cooling water to bearings.
Circulating pump for refrigerating engine	The sea	Condenser of refrigerating engine.
Drinking-water pump	Drinking-water tank	Galley. Drinking water filter.

TABLE No. 33.

Pump Diagram of German Cruiser.

Name of Pump.	Number of Pumps.	Suction.											Delivery.									
		Condensers.	Hot Well.	Feed-water Tank in Double Bottom of Ship.	Fresh-water Tank in Double Bottom of Ship.	Overboard.	The Sea.	Main Bilge Piping.	Auxiliary Bilge Piping.	Bilge Suction to Double Bottom.	From Engine-room Bilge.	Drinking-water Tank.	Hot Well.	Feed-water Tank in Double Bottom of Ship.	Boiler.	Ash Ejector.	The Sea.	Fire Hose.	To Decks.	Fresh-water Tank and Piping.	Sea-water Piping.	Drinking-water Tank and Piping.	Overboard.
Centrifugal circulating pumps	3						×	×									×*						
Centrifugal bilge pumps	1							×									×						
Engine bilge pumps	3										×						×						
Air pumps	3	×											×										
Steam-driven bilge pumps and ash ejectors	4						×		×	×						×	×	×†	×				
	1						×														×		
Drinking-water pumps	1					×						×										×	
Fresh-water pumps	1				×	×														×			
Main feed pumps	4		×												×								
Spare feed pumps	4		×	×		×‡								×‡	×								×‡

* Direct or through the condensers. † To the fire hose, and to wash the decks. ‡ Only the pumps in the two forward engine-rooms.

Lloyd's Rules for Pumps and Pumping Arrangements.

§ 193. RULE No. 20. The engines are to be fitted with two feed pumps, each capable of supplying the boilers; the pumps, &c., to be so arranged that either can be overhauled whilst the other is at work.

21. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled whilst the other is at work.

22. In engines of 70 H.P. and under, one feed pump and one bilge pump will be deemed sufficient, provided they are of adequate capacity. The main feed pumps may be worked by independent engines, provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by paragraph 25 should also be fitted with an automatic speed regulator.

23. A bilge injection, or a bilge suction to the circulating pump, is to be fitted.

24. The engine bilge pumps are to be fitted capable of pumping from each compartment of the vessel. The mud boxes and roses in engine-room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

25. A steam pump is to be provided, capable of supplying the boilers with water, this pump to be provided with suctions to the hot well, and also to the sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and if no hand pump is fitted in engine-room, it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all these requirements.

26. In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining the pipes, and all main steam pipes are to be tested by hydraulic pressure to twice the working pressure, in the presence of the engineer surveyor.

27. All discharge pipes to be, if possible, carried above the deep load line, and to have discharge valves fitted on the plating of the vessel in an accessible position.

28. No pipes to be carried through the bunkers without being properly protected.

29. Bilge suction pipes to be arranged to pump direct from each

compartment, the roses to be fixed in places where they can be easily accessible.

A suction pipe from the bottom of the boiler to the steam pump must be separated from all the other suction pipes belonging to the pump, that the boiler pressure cannot, through carelessness or ignorance, find its way into any of the piping.

. . . . . .

39. Cocks and valves connecting all suction pipes to be fixed above the stoke-hold and engine-room platforms.

40. The arrangement of pumps, bilge injections, suction and delivery pipes is to be such as will not permit of water being run from the sea into the vessel, by an act of carelessness or neglect. Any defective arrangement to be reported to the Committee.

41. In steam vessels the pumping arrangements, according to the division of holds, &c., to be as follows:—

42. *Holds with Double Bottoms.*—In the double bottom of each compartment of the hold and of engine and boiler space, a steam pump suction is to be fitted at the middle line, and one on each side, to clear the tanks of water when the vessel has a heavy list. Where there is considerable rise of floor towards the ends of vessels, the middle line suction only will be required. A steam pump suction and a hand pump are also to be fitted to each bilge, in each hold where there is no well. When there is a well, one or three steam pump suctions are to be fitted in the same, according to the rise of floor, whether considerable or little, and hand pumps are fitted at the bilges.

43. *Holds without Double Bottoms.*—Where there is considerable rise of floor, one steam pump suction and one hand pump are to be fitted in each hold. In vessels with little rise of floor, two or three steam pump suctions and at least one hand pump to be fitted to each hold.

44. *Engine and Boiler Space.*—Where a double bottom extends the whole length of engine and boiler space, two steam pump suctions are to be fitted to the bilge on each side. Where there is a well, one steam pump suction should be fitted in each bilge, and one in the well. Where there is no double bottom in the machinery space, centre and wing steam pump suctions should be fitted. The rose box of the bilge injection is to be fitted where easily accessible, and is to be used for bilge water only. The main and donkey pumps to draw from all compartments, and the donkey to have also a separate bilge suction in the engine-room.

45. *Fore and After Peaks.*—If the peaks are fitted as water ballast tanks, a separate steam pump suction is to be led to each. If not used for water ballast, an efficient pump is to be fitted in the fore peak. If

the after peak is used as a ballast tank, no sluice valve or cock is to be fitted to the after bulkhead; but if it is not so used, and if no pump is fitted in it, a sluice valve or cock is to be fitted to the after bulkhead, to allow water to reach the pumps when required.

46. *Tunnel.*—The tunnel well is to be cleared by a steam pump suction.

47. *All Hand Pumps* to be capable of being worked from the upper or main decks above the deep load water line; the bottoms of the pump chambers are not to be more than 24 feet above the suction rose, and the pumps are to be tested by the surveyors, to ensure that water can be pumped from the limbers. The sizes of the hand pumps to be not less than those given in the following table:—

Tonnage under Upper Deck.	Hand Pumps in Holds.	
	Diameter of Barrel.	Diameter of Tail Pipe.
	Inches.	Inches.
In vessels under 500 tons - - -	4	2
In vessels of 500 tons but under 1000 tons - - - - - - -	4½	2¼
In vessels of 1000 tons but under 2000 tons - - - - - - -	5	2½
In vessels of 2000 tons and above -	5½	2¾

In lieu of hand pumps in each compartment an approved fly wheel pump may be fitted, if it is connected to the steam pump bilge suction pipes of these compartments.

48. *No Sluice Valve or Cock* is to be fitted to the collision bulkhead.

49. *No Sluice Valves or Cocks* are to be fitted to the engine-room or other watertight bulkheads, unless they are arranged so as to be at all times accessible.

50. When *Sluice Valves* are fitted, they must be so arranged as to be controlled above the load water line, and the rods are to be boxed-in to prevent injury.

51. *Sounding Pipes* to be fitted on each side of holds and ballast tanks, and a doubling plate is to be fitted under each.

52. *Air Pipes* to be fitted to each ballast tank as required.

53. All *Cocks* and *Valves* in connection with bilge and ballast suction pipes are to be fitted in places where they are at all times accessible.

54. The *Filling Pipes* for deep tanks, which can be used for either cargo or ballast, must be controlled by valves placed in an accessible position, and so arranged that when the tank is being used for cargo it will be impossible to fill it with water. This result is to be obtained by taking out a short bend or wedge piece, and fitting blank flanges in its place, or in some other way to be submitted to and approved by the Committee.

55. The *Pipes* for bilge or ballast suctions are to be fitted with flanged joints in convenient lengths, so that they may be easily disconnected for clearing. In the case of cast-iron suction pipes, which are not also used as tank-filling pipes, or which cannot be subjected to sea pressure, spigot and faucet joints made with indiarubber rings fitted over the spigots might be adopted, except in the case of bilge suction pipes passing through ballast tanks, which should be fitted with flanged joints.

56. The *Suction Pipes* to fore and aft peaks and to the tunnel well should not be less than $2\frac{1}{4}$ inches inside diameter, except in vessels under 500 tons under deck, in which case they may be made 2 inches.

57. The *Bilge Injection* should not be less than two-thirds of the diameter of the sea inlet to the circulating pump.

58. The inside diameter of other bilge suction pipes should not be less than that given in the following table:—

Tonnage under Upper Deck.	Engine-room Centre Suction, Separate Donkey Suction, and Hold Centre Suctions.	Wing Suctions in Holds where no Centre Suctions are fitted, and Wing Suctions in Engine-room.	Wing Suctions in Holds where Centre Suctions are also fitted.
	Inches.	Inches.	Inches.
In vessels under 500 tons - -	2	2	2
In vessels of 500 tons but under 1000 tons - - - - -	$2\frac{1}{4}$	2	2
In vessels of 1000 tons but under 1500 tons - - - - -	$2\frac{1}{2}$	$2\frac{1}{4}$	2
In vessels of 1500 tons but under 2000 tons - - - - -	3	$2\frac{3}{4}$	$2\frac{1}{4}$
In vessels of 2000 tons but under 3000 tons - - - - -	$3\frac{1}{2}$	3	$2\frac{1}{2}$
In vessels of 3000 tons and above	$3\frac{1}{2}$	$3\frac{1}{2}$	$2\frac{3}{4}$

In cases where more than one suction to any one compartment is connected to the pumps by a single pipe, this pipe should be not less than the size required for the centre suction.

PART III.

SHAFTING, RESISTANCE OF SHIPS, PROPELLERS.

SECTION I.

SHAFTING.

Thrust Shaft and Thrust Block.

§ 194. **Axial Thrust.**—In the engines of all screw steamers the thrust of the propeller is taken up by what is known as a thrust shaft. It consists of collars forged on to the shaft which press against a thrust bearing.

The thrust shaft and thrust block are calculated to withstand the "indicated thrust," that is the axial thrust which would be produced by the propeller, if all the power generated by the engine in I.H.P. were utilised, *without loss*, in driving the ship forward. If I.H.P. is the indicated horse-power of an engine, n the number of revolutions per minute, H the pitch of the screw in feet, P the indicated thrust in pounds, then the equation will be—

$$\frac{\text{I.H.P.}}{n} \times 33{,}000 = \text{P} \times \text{H}.$$

That is, the work done during one revolution of the engine must be equal to the work due to the thrust during one revolution of the screw, acting through a distance equal to the pitch of the screw. From the above equation we get—

$$\text{P} = \frac{\text{I.H.P.} \times 33{,}000}{n\text{H}}$$

If, for instance, the efficiency of the screw is about 65 %, and the mechanical efficiency of the engine about 85 %, then the "effective thrust" will be—

$$0{\cdot}85 \times 0{\cdot}65 \times \text{P} = \text{about } 0{\cdot}55\text{P}.$$

As already stated, however, the dimensions of the thrust shaft and bearing are calculated from the indicated thrust P.

§ 195. **Thrust Shaft.**—In all large ships, with engines above 100 I.H.P., the thrust collars are on a separate length of shaft, which is made as short as possible, the object being to simplify the construction and erection of the thrust shaft, and also the fitting and stowing of the spare thrust shaft. The thrust shaft is therefore only made long enough to accommodate sufficient thrust collars, two bearings and the

nuts of the coupling bolts for the flange couplings. The diameter d of the thrust shaft is generally equal to that of the tunnel shaft (see page 340), but it is better to make it thicker, and equal to that of the crank shaft, as the thrust shaft is more likely to be subjected to bending strain from the crank shaft than the tunnel shaft.

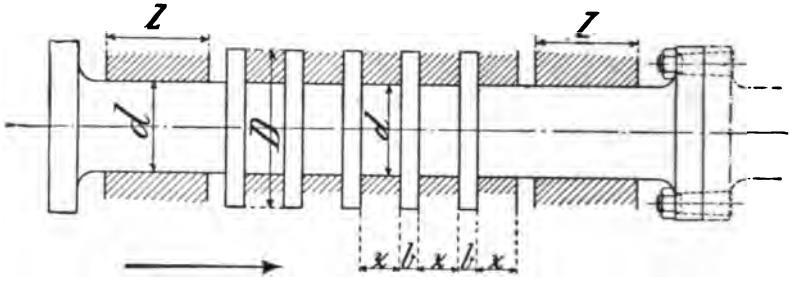


Fig. 317.

Diameter of the collars $D = 1{\cdot}6$ to $1{\cdot}9d$ (Fig. 317).
Width of collars $b = 0{\cdot}13$ „ $0{\cdot}16d$ in lightly built engines, or in the case of strong shafts.
„ $b = 0{\cdot}15$ „ $0{\cdot}2d$ in heavily built or small engines.

For space between the collars z see § 196.

The number of rings is such that the pressure p exerted by the "indicated thrust" upon their "effective area" is as follows:—

$p =$	40 to	55 lb.	per square inch	for cargo steamers.
$p =$	55 „	80	„ „	passenger steamers.
$p =$	70 „	85	„ „	heavy warships.
$p = 100$ „		130	„ „	light warships.

The coupling flanges of the thrust shaft are the same as those for the crank shaft, and the material used in the construction of both shafts is the same.

§ 196. **Thrust Block.**—In large ships the block is generally a square trough of cast iron or cast steel, with bearings at each end (see Figs. 321 to 324). Two heavy screwed rods are fixed one on each side of the shaft, and to these the horse-shoe thrust collars are attached. They are secured and adjusted by means of nuts on the two side rods. The nuts transmit the thrust to the two rods, and through them to the main thrust-block casting, and thence to the ship. In large ships the horse-shoe collars between the thrust collars (Figs. 318 to 320) consist generally of hollow cast-iron or steel castings (the latter usually only in

warships). Their bearing surfaces are faced with white metal fitted or cast on in the ordinary way, and cooling water circulates through the hollow interior. (See also § 267, page 450.) At the top of each horse-shoe an oil cup is fitted, which supplies oil between the thrust collar and the cap. The collars must be strong enough not to bend with the thrust to which each is subjected. When calculating their dimensions it is well to allow very small stresses, as the thrust is often distributed only over a portion of the horse-shoe. The lugs supporting the horse-shoe caps on the side rods should be so arranged, that the line connecting the centres of the rods passes through the

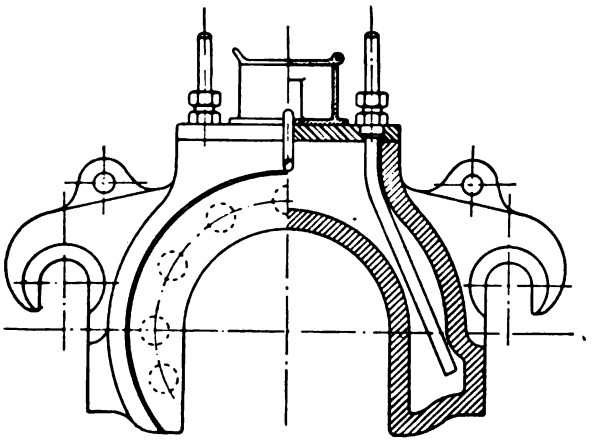


Fig. 318.

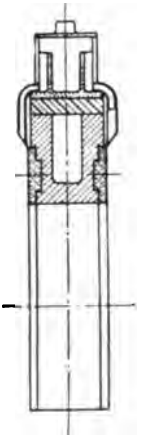


Fig. 319.

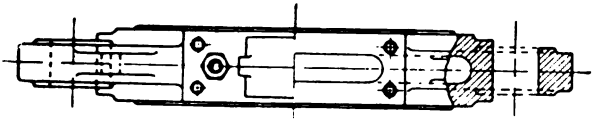


Fig. 320.

centre of gravity of the thrust surface. The latter are usually made horse-shoe shaped in larger engines, as shown in Figs. 318 to 320. In the larger collars there are generally two eye-bolts, to lift them in and out more easily.

Material of the horse-shoe collars—Cast steel or cast iron, less frequently bronze.

Thickness of the horse-shoe collars if solid $z = 2$ to $2{\cdot}5b$ (see Fig. 317).

" " if hollow $z = 2{\cdot}5$ to $3b$ "

Thickness of the white-metal liner $w = \frac{b}{6} + {\cdot}08$ inch.

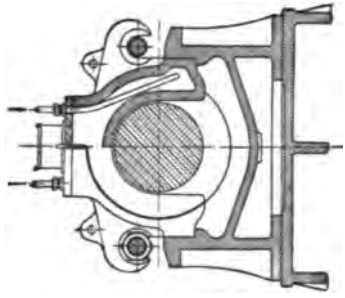


Fig. 322.

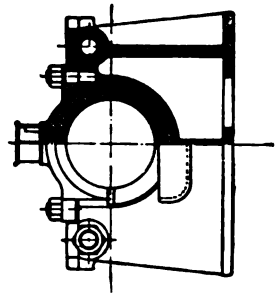


Fig. 324.

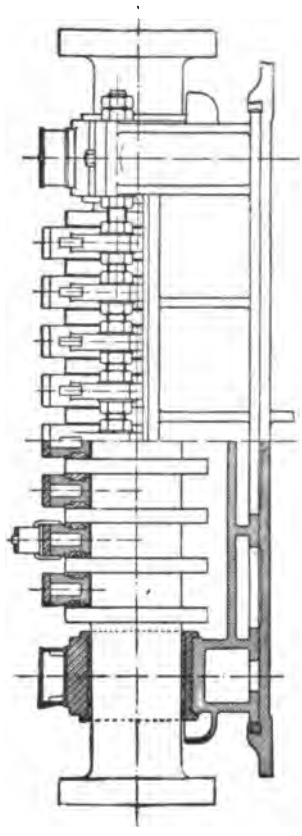


Fig. 321.

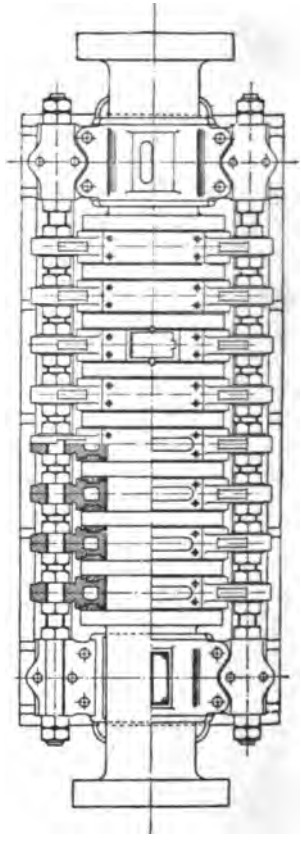


Fig. 323.

The side thrust rods are made of steel, and screwed with a fine thread. The nuts between the horse-shoe caps are made of gunmetal, so that they may not rust on to either the collars or rods. The two side rods are fitted into strong eyes at either end with nuts on each side.

The allowable stress in the side thrust rods at the bottom of the thread, due to the indicated thrust, is $s = 5{,}500$ to $8{,}500$ lb. per square inch. It is desirable to allow the horse-shoe collars to fit against and into the sides of the trough casting, either at the ears, or at the lower part of each shoe. (See Fig. 322.)

Thrust block end bearings:—Length $l = 0{\cdot}8$ to $1{\cdot}2d$ (Fig. 317). Brasses are generally of cast iron or gunmetal, and are lined with white metal.

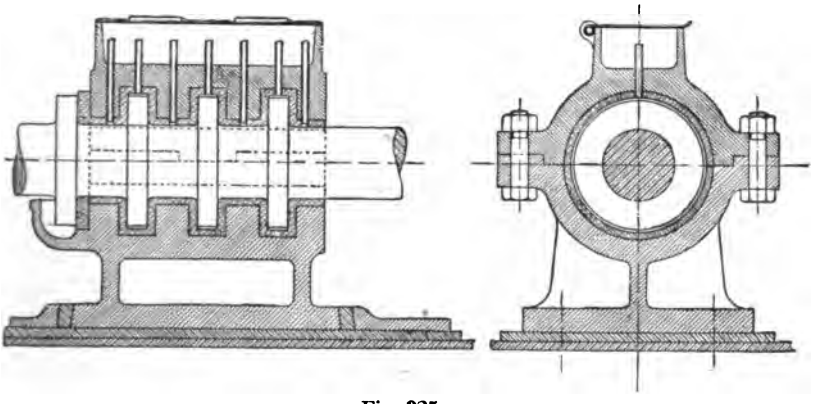


Fig. 325.

The thrust block is secured to the ship as firmly as possible. It is best to connect it to the engine bed-plate, so that part of the thrust may be taken by the foundation bolts. In the engines of large merchant ships the thrust block is frequently placed upon a separate bed-plate, which is connected to the engine bed-plate, and to it the thrust block is secured and fitted with adjusting screws. If the thrust block has no separate independent bed-plate, it should be firmly wedged between cleats strongly riveted to the body of the ship.

§ 197. **Thrust Blocks in Small Ships.**—These have no separate end bearings, and are generally not connected to the engine bed-plate. The thrust is often taken by gunmetal rings, made in two halves, which fit into grooves in the body and cap of the thrust block. The rings must be prevented from moving round. The cap should be stepped

or keyed into the lower part or body of the block. The whole of the latter may be made either of gunmetal or cast iron.

In very small engines the thrust blocks are made of cast iron or gunmetal, and the cap and body only are lined with white metal, in which grooves are turned to receive the thrust collars. In order to transfer the axial thrust upon the cap to the foundation, the cover must, as already mentioned above, be stepped or carefully keyed into the lower part of the block, or the bolts securing it must be an accurate driving fit (Fig. 325).

Tunnel Shafts and Plummer Blocks.

§ 198. **Intermediate or Tunnel Shafts.**—These join on to the after end of the thrust shaft, and transmit the turning moment produced by the engine to the tail shaft. They are therefore exposed to bending and twisting strains, but as they have to take fewer shocks and bending stresses than the crank or propeller shafts, they are generally made smaller than either of these. If d is the diameter of the crank shaft, the diameter of the tunnel shaft will be $d_1 = 0{\cdot}85$ to $1d$.

For *German Lloyd's Rules for Shafts* see pages 345, 346.

The various lengths of the tunnel shaft are as far as possible made the same, and their length is determined by the dimensions and type of the ship. Care must be exercised that the shafts can be easily taken

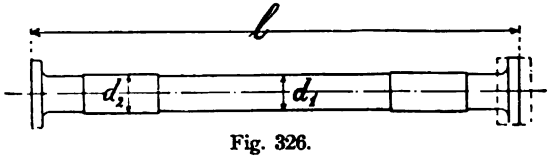


Fig. 326.

in and out and lifted, this being often necessary when overhauling and renewing the propeller shaft. In large ships, where the diameter of the shaft is from 10 to 24 inches (Fig. 326), $l = 16$ to 24 feet. The separate lengths of shaft are joined up, like those of the crank shaft, by flanges. The tunnel shafts are often made hollow, the proportions of the bore being the same as for crank shafts. Close to each coupling is a bearing, each journal (d_2) being made slightly larger in diameter than the body of the shaft to allow for wear.

A flange of one of the intermediate shafts is left very thick, so that, when everything else is in place in the ship (Fig. 326), any differences between the distance from the tail shaft to the thrust block, as originally designed, and as actually built, can be allowed for. The aftermost

intermediate shaft is sometimes provided with a shoulder or collar on each side of the last plummer block, to prevent the tail-end shaft from falling out, in case of an accident to the tunnel shaft.

§ 199. **Plummer Blocks or Bearings.**—Length $l = 0{\cdot}8$ to $1{\cdot}2d_2$. (See Figs. 326 and 327.) Distance of the bearings apart varies according to diameter of the shaft; in large merchant ships and fast steamers there is a plummer block on each side of the shaft couplings. For the length of the shaft see § 198.

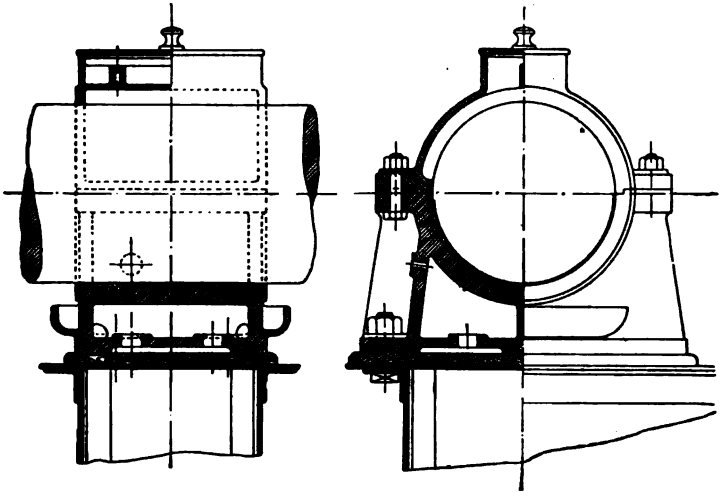


Fig. 327.

Material used for the block—Cast iron with white-metal lining in merchant vessels.

„ „ Cast iron, brass, or cast steel, less frequently gunmetal, with white-metal lining, in warships.

As the plummer block does not have to take any great upward thrust, the lower part only need be strongly built and lined with white metal, the upper part being treated as a protecting cap, and kept quite thin. On the top is a large oil and grease box, to lubricate the bearings with oil and solid grease. In very large engines the plummer block is sometimes made a hollow casting, and cooling water is circulated through it.

§ 200. **Bulkhead Stuffing Boxes.**—At the places where the

shafting passes through the watertight bulkheads, a stuffing box made in two halves has to be provided. It is advisable to cast it in two halves in the form of a flanged plate, so that it can be taken apart, if the shaft

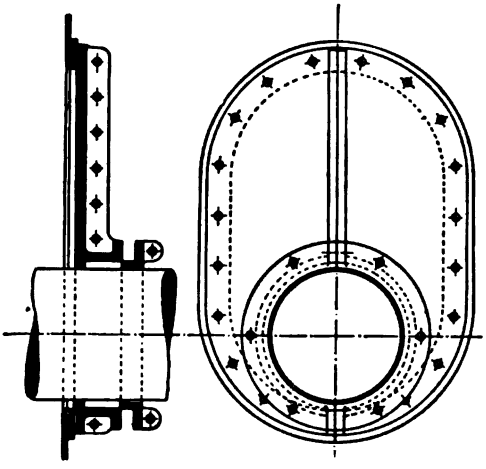


Fig. 328.

has to be lifted and taken out (Fig. 328). For smaller ships the plate on the bulkhead is made of sheet metal, and the stuffing box fixed to it.

§ 201. **Shaft Brake.**—At a suitable place in the tunnel a strong band brake is often placed to lock the shaft. This is used if the latter, for any reason, has to be uncoupled from the engine. It might otherwise be set in motion by the screw, as the ship proceeds.

Shaft Couplings.

§ 202. **Detachable Shaft Couplings.**—These are used to couple together either the tail shaft and the aftermost tunnel shaft, or any two lengths of the aftermost shafting. They are provided in case it should be necessary to draw out the propeller shaft from the stern. As it would be extremely difficult to carry the propeller shaft through the ship when built, and fit it into the stern tube from within, one of the following arrangements is usually fitted in large vessels.

1. In the coupling shown in Fig. 329, the flange, which fits the shaft exactly, is drawn over the end of the shaft by hydraulic pressure, and a strong feather prevents it from turning. To prevent the propeller shaft working forward, should the flange get loose, the diameter of the part of the shaft in the flange is reduced to d_2. To avoid the danger of its working back, the shaft is protected by a ring in two parts, which is fitted into the groove at the forward end, and held fast in position by screwing up the flange against it. To draw off the flange, when the propeller shaft is taken out of the ship in dock, hydraulic pressure may have to be used. Let d be the diameter of the propeller shaft; then

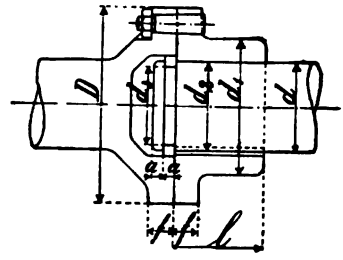


Fig. 329.

Diameter of the boss	$d_1 = 1{\cdot}5d.$
„ „ flange	$D = 2$ to $2{\cdot}5d.$
„ „ bore of the movable flange	$d_2 = d - \frac{d}{30}$ to $d - \frac{d}{60}$
Length of the movable flange	$l = 0{\cdot}8$ to $d.$
Thickness of collar	$a = 0{\cdot}12$ to $0{\cdot}15d.$
Recess for the collar	$d_3 = d - a.$
Thickness of flange	$f = 0{\cdot}25$ to $0{\cdot}3d.$

2. The removable coupling or flange is often secured to the end of the shaft by a cone and nut, and a strong feather or key prevents its turning. Material of the flange, forged or cast steel. Principal dimensions, as above. Diameter of the screw, about 0·5 to 0·6d. Depth of nut, about 0·4d; taper about 1 in 6 to 1 in 10.

§ 203. **Muff Couplings.**—The coupling described under § 202 has the disadvantage that it is very difficult to take to pieces, and if the shaft becomes worn by being often removed and replaced, it does not fit tightly enough. A so-called "muff coupling" (Fig. 330) is therefore frequently used. It consists of two half sleeves of wrought or cast steel, which grip the ends of the two shafts by means of several strong bolts. Each end of the shaft is prevented from turning in the sleeve by a strong key. (For dimensions of the latter see page 401). The two half sleeves are fitted with a little clearance between them, and it is only when the bolts are drawn up tight that they grip the shaft.

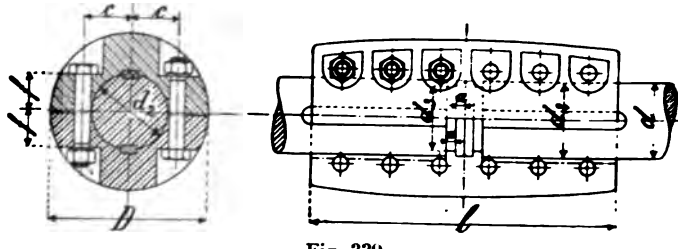


Fig. 330.

Dimensions (see Fig. 330).

Diameter of the propeller shaft	$= d$.
Length of coupling	$l = 3$ to $3{\cdot}8d$.
Bore ,,	$d_2 = d - \frac{d}{20}$ to $d - \frac{d}{40}$
Recess for the collars	$d_3 = d - a$.
External diameter of coupling	$D = 1{\cdot}8$ to $2d$.
Width of collar left at end of shaft and of recess	$a = 0{\cdot}12$ to $0{\cdot}15d$.
Thickness of flange	$f = 0{\cdot}5$ to $0{\cdot}6d$.

Total area of all bolts on both sides at the bottom of the thread—

$$A = 0{\cdot}35 \text{ to } 0{\cdot}5l\left(\frac{D - d_2}{2}\right)$$

The distance c of the bolts from the centre of the shaft is as small as possible; they are frequently allowed to cut slightly into the sides of the shaft. The strength of the bolts should be such that by tightening them up to a given allowable stress, the pressure of the sleeves is sufficient to grip by friction alone.

§ 204. **Disconnecting Couplings.**—These are sometimes used in warships having several engines, and they are placed in the shafting immediately behind the engine, so that if the ship is steaming very

slowly, one of the shafts can be allowed to turn freely without turning the engine. In triple-screw steamers it is sometimes desirable to be able to uncouple either the two outermost or the central shaft. These couplings are made—

I. As claw or clutch couplings.
II. As flange couplings, with circular bolts which can be easily inserted and withdrawn.

As a rule it is desirable, if possible, to avoid such couplings, as even when most carefully made they form a weak point in the shafting, and often begin to knock; this is due to the twisting strain on the shaft, when a certain engine speed is reached. Loose couplings have therefore been much less used of late years in warships.

§ 205. **Tail or Propeller Shaft.**—In small ships this is almost always connected to the tunnel shaft by a solid flange coupling, in larger ships by a detachable coupling (see page 343), so that the tail shaft may be fitted into the ship from the stern. For mode of attach-

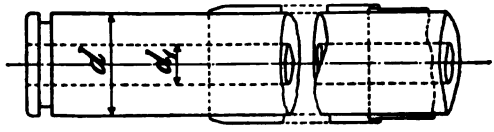


Fig. 331.

ment of the propeller to its shaft see page 401. The length of the propeller shaft is determined by the length of the stern tube, which again depends on the build of the ship. It is advisable not to make the propeller shafts too long, in order that they may the more easily be put in, examined, and if necessary renewed. In ships with fine lines, long tail shafts cannot always be avoided. If, for constructive reasons, the shaft must project considerably beyond the body of the ship, a muff coupling is often placed between the stern tube and the stern bracket.

Diameter of the Tail Shaft.—The dimensions of this are generally greater than those of the crank shaft. Under certain conditions, as when the propeller strikes against a solid object, this shaft is exposed to enormous strains. In any case, it has to withstand great bending strains due to the weight of the overhanging screw, and these are much increased if one of the blades be lost, or the bearings of the stern tube become worn. If d_k is the diameter of the crank shaft, then as a rule $d = 1$ to $1{\cdot}1d_k$.

Extract from the German Lloyd's Rules.—If D is the diameter of the propeller, d must be $= 0{\cdot}6d_k + 0{\cdot}03D$, or at least $= 1{\cdot}02d_k$.

Larger shafts are often made hollow: proportions of the bore are the same as in crank shafts. In all large, and in well-built small, cargo vessels, as well as in all warships, the propeller shaft is generally cased in brass where it runs in lignum vitæ bearings. The brass sleeve is shrunk on while hot, having been previously bored to a diameter slightly less than that of the shaft. Thickness of the brass sleeves for shafts above 10 inches in diameter—

$$\delta = 0{\cdot}60 \text{ inch} + \frac{d \text{ inches}}{50}$$

If the bearings are lined with white metal, the shaft where it runs in them either has no sleeve, or has a nickel-steel sleeve. Between the bearings, the shaft, in high-class work, is protected from corrosion either by a thin brass sleeve shrunk on hot and sweated to the bearing sleeves, or cast in one with them, or by a rubber coating. The part of the shaft in the water, between the stern tube and the propeller, is also frequently coated with rubber in warships, and the rubber protected from injury by a further binding, or by a sheet-metal casing or by both.

Material used for the Propeller Shaft.—*Wrought iron* is now seldom employed; it is not suitable for this part of the shaft because it rusts easily in sea-water and is not strong enough. *Siemens-Martin steel* is used under all ordinary circumstances. Tensile strength, 28 to 32 tons per square inch; extension, 20 to 25 °/₀ on 8 inches. *Crucible steel* is very good for large ships (fast and mail steamers), and is frequently used for warships. Tensile strength is the same as for Siemens-Martin steel. *Nickel steel* is used for the same types of ships as crucible steel. Tensile strength up to 38 tons per square inch; extension, 25 °/₀ on 8 inches.

Stern Tube.

§ 206. **General Remarks.**—The screw shaft, where it comes through the skin of the ship, is enclosed in the stern tube, which forms not only a bearing for the shaft, but a watertight joint round it. The stern tube is a tube fixed at its forward end to one of the aftermost bulkheads, and at its after end to the outer plating. If it is very long, it is often made in two lengths, and supported in the centre. At the inboard and outboard ends, and sometimes also in the middle, long bushes are provided in which the shaft runs. The water is allowed to reach these bushes, and serves to lubricate and keep them cool; in fact an artificial circulation is sometimes produced in the tube, cooling water being either forced into the stern tube or sucked from it by the engine. (See page 450.) Occasionally the stern tube

is closed at its outer end by a stuffing box, and instead of water, oil is forced into the tube, to lubricate the bearings and preserve the shaft.

At the forward or inboard end of the stern tube is a stuffing box, which prevents the water in the tube from finding its way into the tunnel. This stuffing box, which is attached to the cross bulkhead at the after end of the tunnel, must be accessible from the tunnel.

Length of Stern-tube Bearings—

Forward $l_1 = 3$ to $4d$
Aft $l_2 = 4$ to $5{\cdot}5d$

In ships where the screw shaft also runs in a stern bracket beyond the stern tube, the bearings of the latter are generally somewhat shorter.

l_1 = about 2 to $3d$.
l_2 = „ 3 to $4d$.
l_3 = „ 4·5 to $5{\cdot}5d$.

In single-screw ships the propeller shaft is sometimes carried through to a bearing in the stern post.

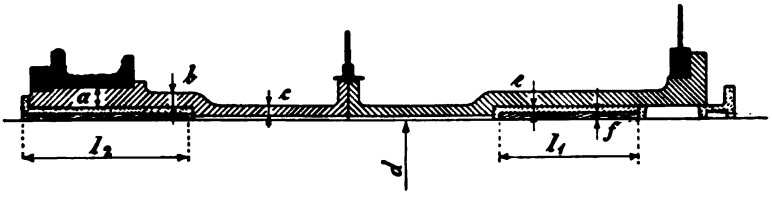


Fig. 332.

Length of Stern Tube.—This depends on the length of the bearings given above, and on the distances between them; the latter depend largely on the form of the ship, and are determined by considerations governing the accessibility of the stern-tube stuffing box at the end of the tunnel. Ships with fine stern lines therefore usually have longer stern tubes than ships with full stern lines. Total length of the stern tube = 15 to $25d$, according to the form of the ship.

Diameter of the Stern-tube Bearings.—If, as is the usual practice, the shaft is fitted in from the stern, it is better to make the diameter of the after bearing somewhat larger than that of the forward one. The reverse holds if the shaft is to be put in from the forward end.

§ 207. **Construction of Stern Tubes for Cargo Boats.**—The stern tube is almost invariably made of cast iron.

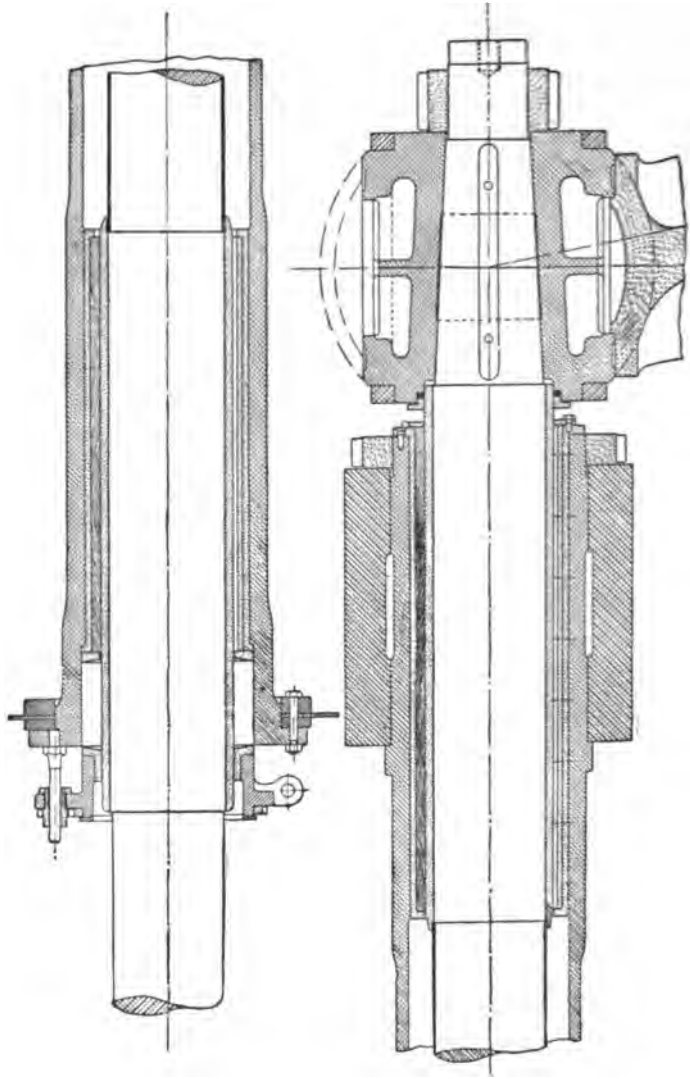


Fig. 333.

If d = diameter of the propeller shaft.

Thickness $c = \frac{d}{20} + 0{\cdot}8$ inch.

$a = 1{\cdot}5$ to $1{\cdot}8c$.

$b = 1{\cdot}2$ to $1{\cdot}5c$.

Removable gunmetal bushes are fitted into the cast-iron tube, which when lined form a bearing for the shaft. (See Fig. 332.) These bushes

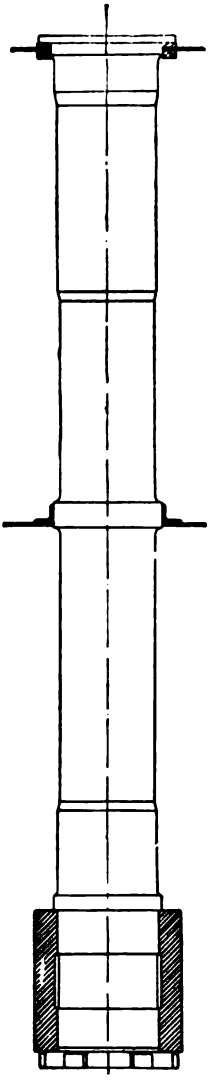


Fig. 334.

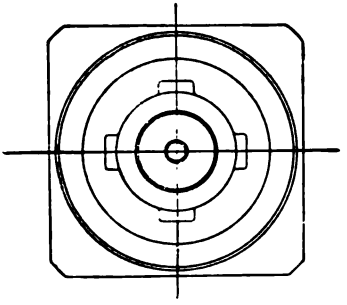


Fig. 335.

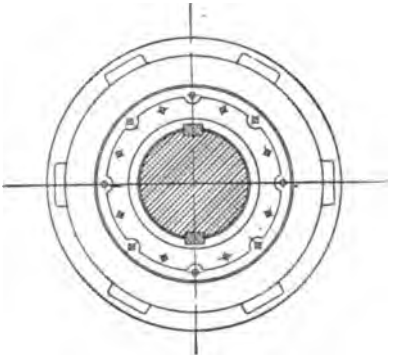


Fig. 336.

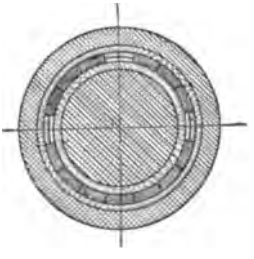


Fig. 337.

are lined with lignum vitæ (see Fig. 337), and for the under side of the bearing it is best to cut the strips across the grain. The lignum vitæ is held in the bushes by longitudinal gunmetal strips, but these should not

be fitted quite at the bottom, so that the shaft may not bear on them. Lignum vitæ is undoubtedly the best material for lining these bushes, and is generally used; it keeps in perfect preservation under water, makes corrosion of the shaft impossible, wears slowly, and is easily renewed. Renewal must, however, take place as soon as the play on the after bearing gets too great; the gunmetal bush can then be drawn out, when the boat is in dock, by means of screws, for which purpose it must be provided on the outside with tapped holes; it is then relined and driven in again by means of a collar and screws. To allow for their expansion longitudinally when wet, the strips of lignum vitæ must not fit tightly at the ends of the bearing.

Thickness of the bronze bushes, $e = \frac{5}{8}$ to 1 inch, for shafts from 8 up to 24 inches in diameter.

Thickness of the lignum vitæ strips, $f = \frac{3}{4}$ to $1\frac{3}{16}$ inches, for shafts from 8 up to 24 inches in diameter.

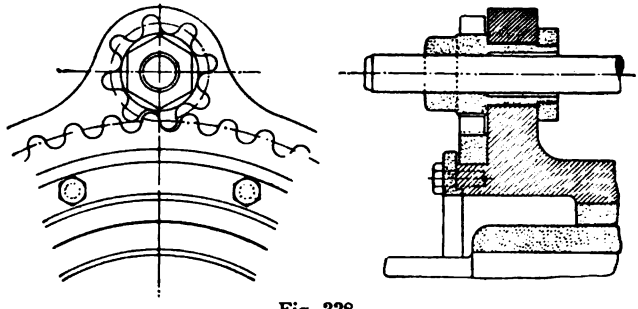


Fig. 338.

In ships working in fresh water, the bearings for the shaft are made in the form of simple cast-iron bushes fitted into the stern tube, and having longitudinal grooves through which the water can circulate. White metal is often used instead of lignum vitæ strips to form the bearing, and occasionally the bush is made entirely of white metal, and drawn into the tube. The composition of the alloy must be determined by its durability in sea water. The thickness of the white-metal bushes, if fitted, is from $\frac{3}{4}$ to $1\frac{1}{16}$ inch.

At the forward or inboard end, the stern tube is bolted to the cross bulkhead at the after end of the screw tunnel by a strong flange, which is strengthened by a stout wrought-iron or steel ring; the after end of the stern tube must be a good tight fit into the stern post, if so fitted. To ensure the hole in the stern post coming in the right place, the stern post must be temporarily bored out, then lined off, and the hole bored afterwards in correct line and position. A thin wrought-iron ring nut, screwed on the end of the stern tube with a fine thread, prevents it from

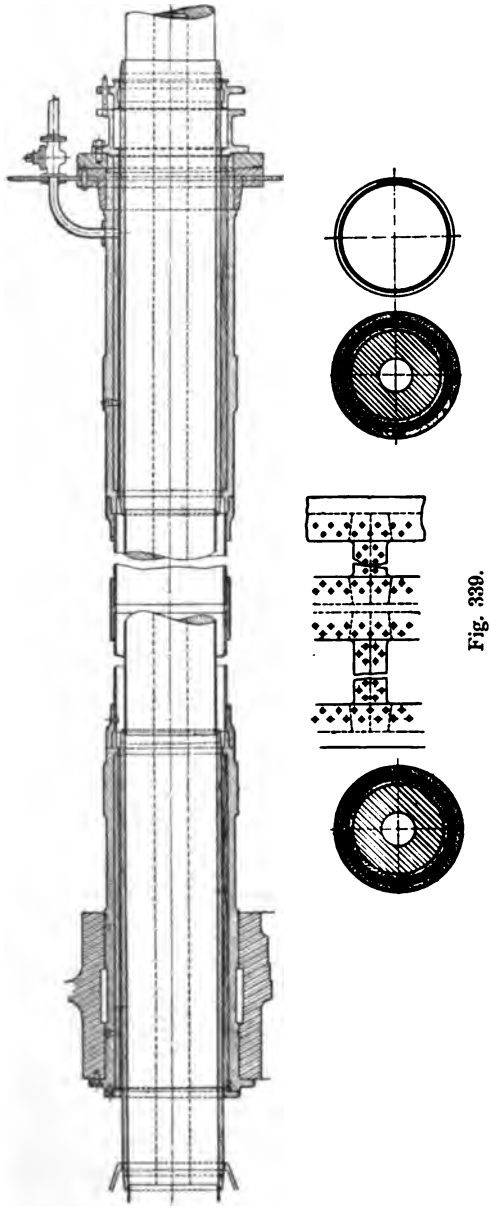


Fig. 339.

being drawn out. Sometimes the stern tube is fitted in from the stern; the flange must then be outside against the stern post, and the nut in front against the bulkhead. This method of construction has the advantage of allowing the stern tube, when the ship is in dock, to be drawn out with ease.

Stern-tube Stuffing Boxes (see Fig. 338).—Width of packing = $\frac{3}{4}$ to $1\frac{1}{2}$ inches, the larger sizes being employed for larger shafts. Depth of packing space = 0·8 to 1·5d, the lower values being employed for larger shafts. In the larger sizes there should be some arrangement for screwing the nuts on evenly. (See Fig. 338.) Cooling water from the engine is sometimes admitted at the after end of the stuffing box.

§ 208. **Method of Construction for Light Warships.**—The stern tube is generally made of brass, and strips of lignum vitæ, or less frequently of white metal, are fitted direct into it. To economise weight, the centre piece is often made as light as possible, and not infrequently consists only of a light metal tube. In torpedo-boats a steel tube is sometimes used for the stern tube, with bronze bushes fitted into it. Brass stern tubes are usually made about half the thickness of cast-iron stern tubes. (See page 348.) For lignum vitæ linings, see page 349.

Fig. 339 shows the stern tube of an armoured cruiser constructed wholly of gunmetal. The centre is composed of plates riveted together, and screwed to the forward and after ends, which are cast in gunmetal, and lined with lignum vitæ.

§ 209. **General Remarks on Shafts.**—The position of the after end of the shafting is determined by the immersion of the propeller in the water, and in twin-screw vessels by this circumstance and the distance of the screws apart. That of the forward end is determined by the position of the engine. To allow for all these conditions the shafting generally slopes downwards towards the stern, and in the case of twin-screw steamers the shafts generally diverge from the centre line as they go towards the stern. In torpedo-boats, where the centre of the screw is deeply submerged in the water, this inclination of the shafting is very marked.

Along the shafting, from the engine-room to the stern-tube stuffing box, there is a passage called *the shaft tunnel*, separated from the adjoining watertight hold of the ship by a roof and watertight bulkheads. It is entered from the engine-room through a watertight door, and has a platform running down its entire length, to make it more accessible. Along the roof of this tunnel, lifting gear is often arranged for the intermediate shafts and their bearings: this gear is used when, in order to examine the propeller shaft and the after stern tube bearing, the tail shaft has to be drawn back into the ship.

SECTION II.

RESISTANCE OF SHIPS.

§ 210. **Froude's Method.**—This is the most accurate method, and is based upon experiments of towing models of ships in a tank, and on what is known as the "Law of Comparison."

The resistance of a ship consists of—

1. Frictional or skin resistance.
2. Eddy resistance.
3. Wave-making resistance.

} Residuary resistance.

Only the two last-named resistances follow the "Law of Comparison," which may be stated as follows:—

Law of Comparison.—Let L be the length, B the breadth, T the draught, V the speed of the ship; further, let $l = \frac{L}{n}$, $b = \frac{B}{n}$, $t = \frac{T}{n}$, and v be the length, breadth, draught, and speed of the model of the ship. Then, if $V^2 = nv^2$, the resistances of the ship and of the model (W and w) will be to each other as their displacements(D and d)—

$$\frac{W}{w} = \frac{D}{d} = n^3 \text{ *}$$

Application of the "Law of Comparison."—Plot the results of towing experiments on the model, as indicated by the curve AAA, Fig. 340. This gives the resistance of the model at various speeds. Next, calculate the frictional resistance of the model, and deduct this from the total resistance AAA; the curve BBB will be produced, representing the residuary resistance of the model. If the scale be changed, in accordance with the law of comparison, the curve BBB gives the residuary resistance of the *ship*. The frictional resistance of the ship must now

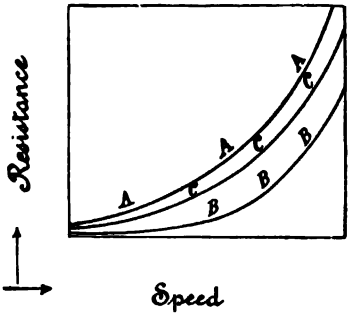


Fig. 340.

* For deductions from this law see Taylor, "Resistance of Ships and Screw Propulsion."

be calculated, and added to the curve BBB, and the curve CCC is thus obtained, which shows the resistance of the ship, if read to the correct or ship scale (Fig. 340).

The *calculation of the frictional resistance* of the model and of the ship is made as follows:—Let W_2 be the frictional resistance in pounds, F * the wetted surface of the ship or model in square feet, v_m the speed of the same in feet per second, γ density of water, f frictional (or skin) resistance at unit speed, in pounds per unit of surface immersed in fresh water, n an index or power, then

$$W_2 = f \times \gamma \times F \times v_m^n$$

The values of f and n can be inserted in the above equation for determining the frictional resistance of the ship, by using the value given below in Table No. 34. For calculating the friction of the paraffin wax model in the water of the experimental tank, somewhat higher values for n and lower for f than those given in the table should be used.

TABLE No. 34.

Constants for Frictional or Skin Resistance of Ships.

(Compare Johow, *Hilfsbuch für den Schiffbau.*)

Length of the Ship on the Water-line.	Ship's Bottom of Iron, well Painted.		Ship's Bottom of Copper or Zinc Sheathing.		Old Foul Ship's Bottom of Copper or Zinc Sheathing.	
Feet.	f	n	f	n	f	n
16·4	0·00026	1·8507	0·00024	1·9015	0·00034	1·8660
32·8	0·00024	1·8427	0·00023	1·8525	0·00031	1·8525
65·6	0·00023	1·8290	0·00023	1·8270	0·00030	1·8430
98·4	0·00023	1·8290	0·00023	1·8270	0·00029	1·8430
131·2	0·00023	1·8290	0·00022	1·8270	0·00029	1·8430
164·0	0·00022	1·8290	0·00022	1·8270	0·00028	1·8430
196·8	0·00022	1·8290	0·00022	1·8270	0·00028	1·8430
230	0·00022	1·8290	0·00022	1·8270	0·00028	1·8430
262·5	0·00022	1·8290	0·00022	1·8270	0·00028	1·8430
295·3	0·00022	1·8290	0·00022	1·8270	0·00027	1·8430
328·1	0·00021	1·8290	0·00022	1·8270	0·00027	1·8430
361·0	0·00021	1·8290	0·00022	1·8270	0·00027	1·8430
393·7	0·00021	1·8290	0·00022	1·8270	0·00027	1·8430

* For calculation of F see page 356.

The values of f do not decrease appreciably—

In Column I. for ships of 550 feet and upwards.
" II. " 425 " "
" III. " 550 " "

§ 211. **Calculation of the Resistance of Ships, and Power required for the Engines of Screw Steamers** (from Middendorf*).—Let

L denote length of the ship on the water-line in feet.
B " greatest breadth of immersed midship section in feet.
T " draught of the ship, excluding keel, in feet.
A " area of immersed midship section in square feet.
F " immersed surface of the ship in square feet.
f " area of propeller disc in square feet.
V_k " speed of ship in knots.
V_m " speed of ship in feet per second.
W " total resistance of the ship in pounds.
W_2 " frictional resistance of the ship in pounds.
W_1 " residuary resistance $W_1 = W - W_2$.
B.H.P. " effective or brake horse-power of the engine.
I.H.P. " indicated horse-power of the engine.
ζ " a coefficient (see Table No. 35).
ϵ " a coefficient (see Table No. 36).
η " efficiency (see Table No. 1). Then

$$1.\ W_1 = \epsilon \times \frac{A \times B \times V_m^{2\cdot5}}{\sqrt{B^2 + \zeta L^2}}$$

$$2.\ W_2 = \cdot00364 \times F \times V_m^{1\cdot85}$$

$$3.\ W = W_1 + W_2.$$

The effective horse-power of the engine is the product of the resist-

* Compare Middendorf, *Schiffswiderstand und Maschinenleistung* (*Jahrbuch der Schiffbautechnischen Gesellschaft*, vol. i., 1900).

ance of the ship and speed of the screw. As v_m is the actual speed of the ship, Middendorf, to estimate the effective horse-power, has increased this factor, so that

$$\text{B.H.P.} = W\left(\frac{v_m}{550} + \frac{1}{167{\cdot}5}\sqrt{\frac{W}{32{\cdot}8}}\right) \text{ and } \text{I.H.P.} = \frac{1}{\eta}\,\text{B.H.P.}$$ (see Table I.)

The values of ζ and ϵ are given in the following tables:—

TABLE No. 35.

Values of the Coefficient ζ.

$\frac{L}{B}$	ζ	$\frac{L}{B}$	ζ	$\frac{L}{B}$	ζ
below 8·5	2·00	9·3 and below 9·4	1·79	10·2 and below 10·3	1·41
8·5 and below 8·6	1·99	9·4 ,, 9·5	1·75	10·3 ,, 10·4	1·38
8·6 ,, 8·7	1·98	9·5 ,, 9·6	1·71	10·4 ,, 10·5	1·35
8·7 ,, 8·8	1·97	9·6 ,, 9·7	1·67	10·5 ,, 10·6	1·32
8·8 ,, 8·9	1·95	9·7 ,, 9·8	1·62	10·6 ,, 10·7	1·29
8·9 ,, 9·0	1·92	9·8 ,, 9·9	1·58	10·7 ,, 10·8	1·27
9·0 ,, 9·1	1·89	9·9 ,, 10·0	1·54	10·8 ,, 10·9	1·25
9·1 ,, 9·2	1·86	10·0 ,, 10·1	1·50	10·9 ,, 11·0	1·24
9·2 ,, 9·3	1·83	10·1 ,, 10·2	1·45	11·0 and over	1·23

The value of η (see Table I.) should always be taken as representing the horse-power of the engine corresponding to the greatest speed. If the speed for a given horse-power of engine has to be calculated before the lines of the ship are got out, F may be approximately determined from the following equation—

$$F = \xi.U.L.$$

Where U = wetted perimeter of midship section in feet.

L = length of the ship in feet.

ξ = a coefficient (see Table No. 37).

TABLE No. 36.

Values of the Coefficient ϵ.

$\frac{L}{V_m^2}$	ϵ	$\frac{L}{V_m^2}$	ϵ	$\frac{L}{V_m^2}$	ϵ	$\frac{L}{V_m^2}$	ϵ
0·10	·255	0·28	·214	0·52	·168	0·88	·122
0·11	·252	0·29	·211	0·54	·165	0·90	·120
0·12	·250	0·30	·209	0·56	·162	0·92	·118
0·13	·247	0·31	·207	0·58	·158	0·94	·117
0·14	·245	0·32	·205	0·60	·155	0·96	·115
0·15	·242	0·33	·203	0·62	·152	0·98	·114
0·16	·240	0·34	·201	0·64	·150	1·00	·113
0·17	·237	0·35	·198	0·66	·147	1·02	·112
0·18	·235	0·36	·196	0·68	·144	1·04	·111
0·19	·232	0·37	·194	0·70	·142	1·06	·110
0·20	·230	0·38	·193	0·72	·139	1·08	·109
0·21	·228	0·39	·191	0·74	·136	1·10	·108
0·22	·226	0·40	·189	0·76	·134	1·12	·108
0·23	·224	0·42	·185	0·78	·131	1·14	·107
0·24	·222	0·44	·182	0·80	·129	1·16	·107
0·25	·220	0·46	·178	0·82	·127	1·18	·106
0·26	·218	0·48	·174	0·84	·126	1·20	·105
0·27	·216	0·50	·171	0·86	·124	and over	

TABLE No. 37.

Values of ξ.

v Speed in Knots.	ξ	v Speed in Knots.	ξ
8 and under 9	0·90	19 and under 20	0·79
9 „ „ 10	0·89	20 „ „ 21	0·78
10 „ „ 11	0·88	21 „ „ 22	0·77
11 „ „ 12	0·87	22 „ „ 23	0·76
12 „ „ 13	0·86	23 „ „ 24	0·75
13 „ „ 14	0·85	24 „ „ 25	0·74
14 „ „ 15	0·84	25 „ „ 26	0·73
15 „ „ 16	0·83	26 „ „ 27	0·72
16 „ „ 17	0·82	27 „ „ 28	0·71
17 „ „ 18	0·81	28 „ „ 29	0·70
18 „ „ 19	0·80	29 „ „ 30	0·69

TABLES

Dimensions, Resistance, and

(Compiled from the Complete Tables of Middendorf, *Jahrbuch*

Type of Ship.	Fast Steamers.				Slower Running Large Cargo and Passenger Steamers.		
Name of Ship.	Fürst Bismarck.	Kaiserin Maria Theresa.	Kaiser Wilhelm d. Grosse.	Lahn.	Aachen, Halle, &c.	Barbarossa, Bremen, &c.	Pensylvania, Pretoria.
Length L in feet	502·5	527·45	625	450.1	355·1	533	561
Breadth B in feet	57·5	51·83	66	48·82	43·5	60	62
Area of midship section A, sq. ft.	1226·6	1143·7	1682	1111	843·5	1477	1500
Immersed surface F, sq. ft.	35,830	39,490	54,876	30,235	24,940	46,440	49,722
Area of propeller disc f, sq. ft.	564·68	530·0	714·1	388·4	228·1	460	427·7
Number of propellers	2	2	2	1	1	2	2
Speed of ship v_k in knots	20·7	20·5	22	18	12	14·4	14
Speed of ship v_m in feet per second	35	34·62	37·15	30·33	20·26	24·31	23·67
Coefficients ζ	1·97	1·45	1·75	1·83	2·0	2·0	1·89
Coefficients ϵ	·105	·105	·105	·105	·105	·105	·105
Coefficients $\frac{1}{\eta}$	1·10	1·10	1·10	1·10	1·21	1·14	1·15
Resistance of ship w_1 in lbs.	75,665	68,965	118,240	47,529	14,129	36,626	34,322
Resistance of ship w_2 in lbs.	93,504	101,205	160,270	60,920	22,784	61,983	62,930
Resistance of ship $w = w_1 + w_2$ lbs.	169,169	170,170	278,510	108,449	36,913	98,609	97,252
Indicated horse-power = I.H.P. calculated	15,189	15,287	27,005	8,672	2,238	6,688	6,574
Indicated horse-power = I.H.P. on trial	15,725	17,260	26,630	8,465	1,775	6,371	5,232 at about 13·2 knots.

Nos. 38 and 39.

Horse-power of various Ships.

der Schiffbautechnischen Gesellschaft, vol. i., 1900.)

	Large Warships.				Torpedo-boats.			Steam Launches and Pinnaces.	Steam Trawlers.
Prinz Henrich, Prinz Regent Luitpold.	Imp. Yacht Hohenzollern.	Minneapolis.	Powerful and Terrible.		Cushing.	Rodgers.	Turbinia.	—	Dora.
455·5	382·5	412	500		138	160	108·24	52·5	105
51	46	58	71		14·24	16	9	10·23	21
1108	736	1124·4	1646·2		47·7	62	23·2	23·45	105·4
33,786	21,272	27,976	40,403		1804·4	2446·8	968·4	592	2651·2
357·6	342·25	518·2	596·7		13·88	102·2	16	8·44	48·67
2	2	3	2		1	2	9	1	1
13·6	21·5	23·1	24·0	21·9	22·5	24·9	32·5	12·5	10·5
22·97	36·31	39·01	40·52	36·98	38·0	42·05	54·89	21·10	17·73
1·92	2·0	2·0	2·0	2·0	1·67	1·54	1·23	2·0	2
·105	·105	·105	·113	·105	·207	·21	·252	·192	·105
1·22	1·12	1·10	1·10	1·10	1·29	1·35	1·2	1·65	1·61
23,701	51,957	111,222	195,173	143,740	7,004	12,005	9,768	1,227	2,100
40,529	59,538	89,420	138,613	117,006	5,492	9,029	5,814	608	1,970
64,230	111,495	200,642	333,786	260,746	12,496	21,034	15,582	1,835	4,070
4,368	10,595	20,184	36,115	25,542	1,618	2,597	2,426	162·7	278
4,280	9,502	20,088	—	25,930	1,730	2,379	2,071 at about 31·7 knots.	177	295

§ 212. **Approximate Method for Determining the Horse-power of an Engine.**—To calculate the work of the engine, approximately and rapidly, for a ship of given deplacement D and speed V, the following formula may be used—

$$\text{I.H.P.} = \frac{V^3 \times D^{\frac{2}{3}}}{C}$$

C is here a constant, namely—

For large and fast ships and steamers	C = 0·00275.	350
For large cargo steamers with full lines	C = 0·00390.	250
For medium-sized warships with fine lines	C = 0·004 to ·005.	250-200
For small ships with fine lines	C = 0·005 to ·006.	200-150

SECTION III.

THE SCREW PROPELLER.

§ 213. **Introduction.**—With our present experience it is not yet possible to draw up simple formulæ which can be universally applied to determine the leading proportions of screw propellers, or from which, if given values are inserted in them, the required proportions can be deduced without difficulty. It must be stated at the outset that no method of calculation can be accepted, unless the results are compared with similar data from actual practice, or checked at least by a second method. The following are the symbols most frequently used:—

I.H.P., indicated horse-power.
B.H.P., brake or effective horse-power.
S.H.P., effective or useful horse-power delivered at the screw propeller.
n, revolutions per minute.
V, speed of the ship in knots.
C, theoretical speed of the screw in knots $= \frac{n \times H \times 60}{6{,}086{\cdot}44}$
U, ,, ,, "wake" in knots.
S_t, real slip in knots.
S_s, apparent slip in knots.
W, total resistance of the ship.
D, diameter of the screw in feet.
H, pitch ,, ,,
d_n, diameter of the boss in feet.
d, diameter of any given element of the blade in feet, and sometimes also
d, diameter of propeller shaft in feet.
z, number of blades.
A, developed area of *one* of the blades in square feet.
l, actual breadth of a blade at any given point in feet (a function of the developed area).
l_m, mean width of blade $= \frac{A}{\frac{D - d_n}{2}}$ in feet.

b, ratio of mean width of blade to diameter of screw—

$$b = \frac{l_m}{D} = \frac{A}{\frac{D(D - d_n)}{2}}$$

k, ratio of total developed area of screw to the disc area $= \frac{z \cdot A}{\frac{\pi}{4} D^2}$

h, thickness of blade (*i.e.*, greatest thickness at any given cross section of blade).

h_o, thickness of blade at tip.

h_n, ,, ,, root.

κ_1 and κ_2, coefficients of the screw.

k_1 ,, k_2, ,, for calculating the stresses in the blades.

§ 214. **General Remarks.**—Every screw consists of a boss and blades. The after surface of the blade, by which the water is forced astern as the ship is driven ahead, is called its "*face.*" It almost always forms a part of the helical surface of an ordinary screw. (Compare page 395.) This surface is produced by a straight line rotating at uniform speed round an axis which it intersects, while the point of intersection also moves along the axis at uniform speed, the angle between this line and the axis remaining constant. That part of the surface of the screw which is required for the blade is bounded by the shape or contour of the blade. The surface of each blade belongs to its own particular part of the screw, and the surfaces of all the blades are parallel to each other. A two-bladed screw is thus two-threaded, a three-bladed screw three-threaded, and so forth.

A distinction is made between *right-handed* and *left-handed* screws. To an observer looking at a screw propeller from aft, a right-handed screw, when working ahead, would appear to revolve in the same direction as the hands of a clock. This motion is said to be "clockwise," and the opposite motion to be "counter-clockwise."

The material required to give the necessary strength to the blade is placed on the back of the driving surface (*i.e.*, the surface next the hull of the ship). (Compare page 396.) The forward surface of the blade does not conform truly in practice to the surface of a true screw. This is more marked nearer the boss, where the blade is thickest.

The edge of the blade which, as the ship moves forward, first cuts the water, is called the "*leading edge*," and the opposite edge is called the "*following edge.*"

The diameter of the screw D is the diameter of the circle described by the tips of the blades.

The *pitch of the screw* H is the distance through which each point on the surface of the blade travels in the direction of the shaft during

one revolution, if the screw were considered as rotating in a fixed solid body.

The ratio $\frac{d}{H}$ is the ratio of the diameter of the circle described by any point on the surface of the blade about the axis, to the pitch. This varies, therefore, for the points of the blade situated at varying distances from the centre, while all the points which lie along a line AB (Fig. 341) of the developed area corresponding to a fixed distance from the axis have the same ratio $\frac{d}{H}$. The *maximum ratio* $\frac{D}{H}$ has on the contrary a definite value for each blade.

The *width of the blade l* (Fig. 341) means the width at any given point on the developed blade, *i.e.*, the actual developed length of the face on the cylindrical section concentric to the shaft. (For length of the cross-sectional template see page 396.)

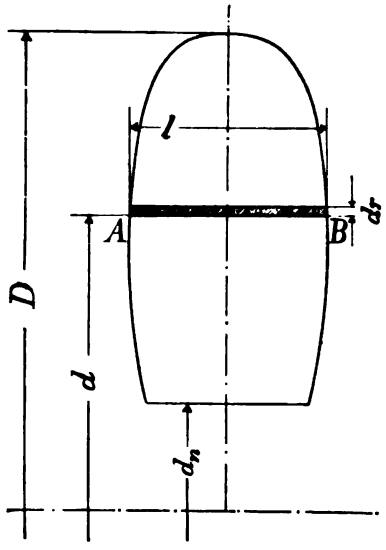


Fig. 341.

The *mean width* l_m is the quotient—

$$l_m = \frac{\text{developed area}}{\text{length of blade}} = \frac{A}{\frac{D - d_n}{2}}$$

Mean width ratio b is the quotient—

$$b = \frac{\text{mean width}}{\text{diameter}} = \frac{l_m}{D}$$

By the term *projected area* of a blade is generally understood the area which is obtained when the blade is projected on a plane at right angles to the axis of the shaft. (Compare Figs. 342 and 346.)

The *developed area* A of the blade (that is the area of its face) can only be approximately determined, because the area of the screw cannot be accurately developed.

To Develop the Area of a Screw: First Method.—Given the projected area, the face of the blade is divided up by any number of concentric circles, the centre of which is the centre of the shaft. These circles cut the projected area (Fig 342) in curves which are arcs of circles, *e.g.*, DAE, and the actual face of the blade in helical lines. The inclination of the latter at A, *i.e.*, the angle which the tangents to

the helical lines at A form with the plane vertical to the centre line of the shaft (the centre of the circles), is obtained from Fig. 344. If r is the radius of any circle cutting the area of the face, H the pitch of the screw which is constant, the angle of inclination of the helical lines with the plane at right angles to the axis is, at every point (and therefore at A), $\tan \alpha = \frac{H}{2r\pi}$. The tangent to the helical lines is thus inclined to the centre of the shaft as the line $B_0O_0C_0$ is to the line $B_0'O_0C_0'$.

Fig. 342.

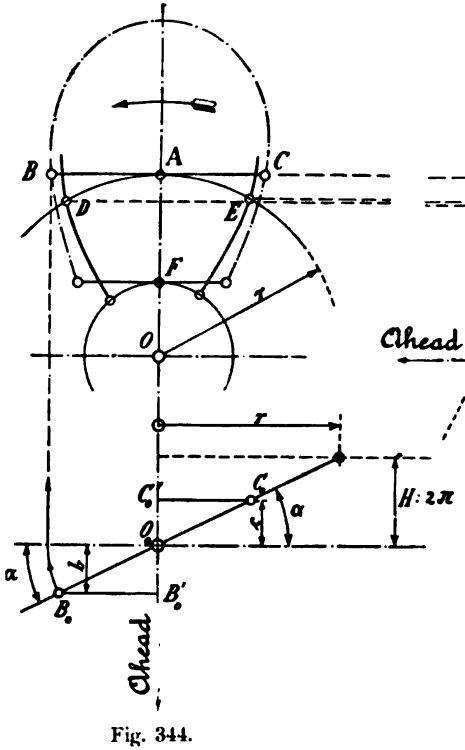


Fig. 343.

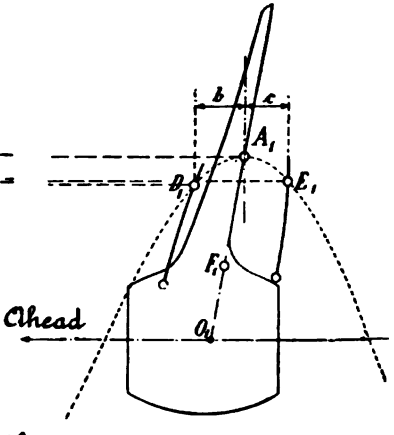


Fig. 344.

Now let B_0B_0' = length of the arc AD, and C_0C_0' = length of the arc AE. Then the true length of the part of the helical line AD = O_0B_0, and the true length of the part of the helical line AE = O_0C_0.

If we make AB at right angles to AO = O_0B_0, and AC at right angles to AO = O_0C_0, then the points B and C will be the points of the developed area corresponding to points D and E of the projected area.

In the same way, as many points on the developed area as are desired may be obtained by determining the true length of the lines of intersection of other circles, concentric to DAE. The innermost of these concentric circles passes through point F, where the centre line of the blade touches the boss. Thus the line at right angles to AO passing through F forms the boundary of the developed area at the boss.

To obtain a *side view* of the blade (Fig. 343) the point A_1 is first determined upon the line $O_1F_1A_1$ (which is here drawn sloping backwards), the distance $b = O_oB_o'$ is plotted out towards the face, and $c = O_oC_o'$ towards the back of the blade, measured from A_1. The points where the horizontal and projected lines through D and E intersect the vertical lines drawn at distances b and c from A_1 give the points D_1 and E_1 of the side view of the blade. From the sectional elevation at the axis and the side elevation of the blade, it is easy to obtain the plan (looking down on the tip of blade). The line $BC = B_oC_o$ is the length of the so-called "cross-sectional template." (Compare Figs. 361, 363, 369, and § 231.) In the side elevation (Fig. 343) the line $D_1A_1E_1$ is thus projected as a sine curve.

If the *developed area* be given, and it is desired to construct the projected area from it, horizontal lines are drawn through the developed area, the angle of inclination α of the helix through A is determined with the help of Fig. 344, $B_oO_o = AB$, $C_oO_o = AC$ are obtained, and B_oB_o' and C_oC_o' drawn at right angles to O_oB_o' and O_oC_o' respectively. If the lines B_oB_o' and C_oC_o' are measured from the point A along the arc of a circle having the radius $r = OA$ described about O, they will give the required points D and E of the projected area.

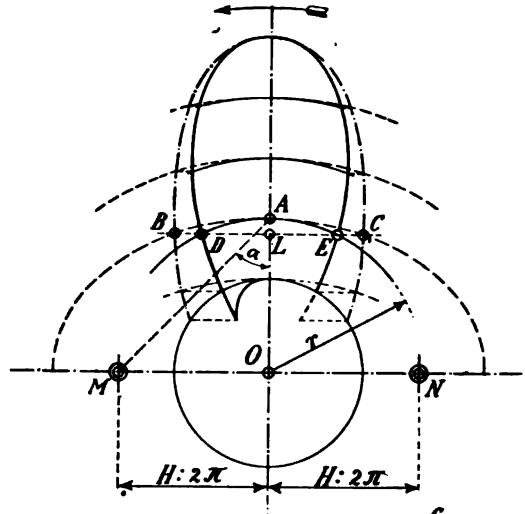
Second Method (see Figs. 345 and 346).—Given the projected area, to find the developed area.

Let D and E be, as before, points on the projected area lying along the arc DAE of a circle with radius r. We will here assume that the area of the screw in the vicinity of point A is represented by the surface of its tangential plane at this point, which closely corresponds to it. The angle of inclination of this tangential plane to the plane of the drawing is equal to the angle of inclination α of the helix of the screw * passing through point A, and can be found from a right-angled triangle with its side and base equal to H and $2\pi r$ or to $\frac{H}{2\pi}$ and r respectively (see Fig. 346). The intersection of the tangential plane

* Strictly speaking this only applies to blades whose planes are vertical to the shaft.

with the circle through A is thus an ellipse, which is projected as a circle DAE. Half the minor axis of this ellipse is equal to the radius r, half the major axis is inclined at the angle α to the plane of the drawing, and when projected is equal to the length r. The major

Fig. 345.



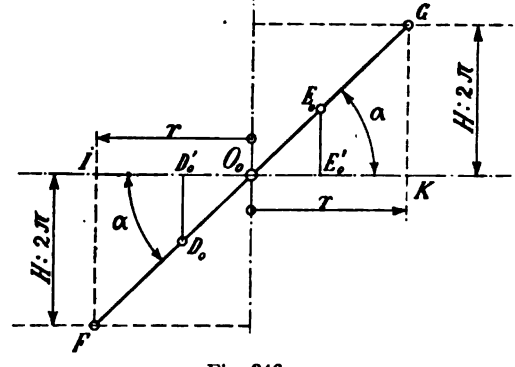


Fig. 346.

axis is thus the hypotenuse of a right-angled triangle, the sides of which are

$$r = O_0I, \text{ and } \frac{H}{2\pi}$$

The focal points M and N of an ellipse may, as we know, be obtained, if from the end A of the minor axis a circle be described with half the

major axis as radius. This has been done in Fig. 345. Taking half the major axis $O_0F = AM$ as radius, a circle is drawn from A, which cuts the line at right angles to AO at M and N, the focal points of the ellipse, the half axes

$$O_0F = AM = AN \text{ ; and } r = OA.$$

As the triangle OAM = triangle IO_0F, then $OM = IF = \frac{H}{2\pi}$. The focal point of the ellipse is thus at the distance $\frac{H}{2\pi}$ from O, and is independent of r. It is therefore the same for all ellipses produced by the intersection of all circles concentric to DAE, with their corresponding tangential planes, and the helical lines of the screw, at the points where the circles intersect the area of the blade. If the tangential plane be turned round into the plane of the drawing, the points in the developed area, corresponding to D and E, namely B and C, are obtained. They are found either by cutting the ellipse passing through any point A by a straight line projected through D and E, which is easily done by means of the focal points M and N, or by the following equation—

$$O_0D_0' = LD \qquad LB = O_0D_0$$
$$\text{and } O_0E_0' = LE \qquad LC = O_0E_0$$

If the *developed area* be given, and the projected area be required, the ellipse BAC is drawn for instance through any point B (with the aid of points M and N), or point A is determined by calculation as follows:—The sum of MB + BN is determined, and MA is made = ½ (MB + BN). Through OA draw a circle having the radius $OA = r$, and find the point where it is intersected by the straight line BLC at right angles to OA. The points of intersection D and E are the required points of the projected area.

§ 215. **Number of Blades.**—*Two blades* are now scarcely ever used, as they strain the shaft, and are particularly prone to give rise to vibrations in the stern of the ship, especially if either of the blades gets damaged. *Three blades* are the almost invariable rule in small and medium sized warships and generally for all small fast vessels. *Four blades* are always employed in ordinary cargo boats, steam tugs, ice-breakers, &c. All other ships have *three* or *four* blades; the larger number is now usually preferred in large fast steamers and passenger vessels. In ships with three screws, the middle screw is sometimes four-bladed and the two outer screws three-bladed.

§ 216. **Different Forms of Blade.**—Blades are very often made which are set over towards the stern (see Fig. 355). This shape

is usually selected, in order that the tips of the blades may (with a single screw) lie well away from the stern-post. These blades are designed to keep the stream of water together, as it is forced astern. For this reason the blades of screws which run at very high speeds are frequently set backwards at a considerable angle, and often have to take a severe stress, as explained in § 226. Blades with variable pitch are seldom used, and have as a rule no advantage over those with uniform pitch. A distinction must be made between blades the pitch of which varies axially, and those in which the pitch varies radially. Torpedo-boats are fitted with screws having an axially, and at the same time a radially varying pitch.

The Hirsch, Mangin, and other similar types of propeller are now seldom fitted.

§ 217. **Speed of the Screw, Stream-line Wake, and Slip.**—If the water were a solid body, the screw would move forward the length of the pitch H at each revolution. The speed of the ship would then be $c = \frac{n \times H}{60}$ feet per second $= \frac{n \times H \times 60}{6{,}086{\cdot}44}$ knots, say $= \frac{n \times H \times 60}{6{,}080}$. The speed c is often called the "speed of the screw." In reality the screw does not, during one revolution, move forward through the distance of the pitch H, but through a smaller distance. The real speed of the ship is therefore less than c. The ratio

$$s_s = \frac{\text{speed of the screw} - \text{speed of the ship}}{\text{speed of the screw}} \quad \text{or} \quad s_s = \frac{c - v}{c}$$

affords a means of estimating the retardation of the screw in the water. The value s_s is called the *apparent slip*. The water in the rear of the ship in which the screw works is not at rest, but follows to some extent the forward motion of the ship. The speed with which the water streams after the body of the ship is called the speed of the *stream-line wake* = U. This is greatest in the part of the water immediately under the stern-post, and round the wetted skin adjoining, and is also greater near the surface of the water than deeper down.*

The value of U, as used in calculations, is the mean speed of the stream-line wake in the region where the propeller works. The "fuller" the stern of a ship, the larger will be the stream-line wake. The effect of the wake upon the screw is the greater, when the latter works in the strongest portion of the wake current. It is therefore

* Compare Calvert "On the Measurements of Wake Currents" (*Inst. of Naval Architects*, 1893).

more marked in single-screw than in twin-screw steamers. In default of other basis of calculation, the diagram given in Fig. 347 may be used to determine U.*

By *fulness* in this diagram is meant displacement ÷ (length × breadth × depth) of the immersed portion of the ship. It is best to calculate U from the results of actual trial trips, and to use the data thus obtained as the basis of new designs.† (Compare page 381.)

A current or wake of a certain velocity is always present. The screw therefore has to impart to the ship only the difference between the velocity of the ship V and the velocity of the wake U. The actual speed of the ship, relatively to the water at the stern of the vessel, is thus V - U. As this value is always less than V, the screw must at

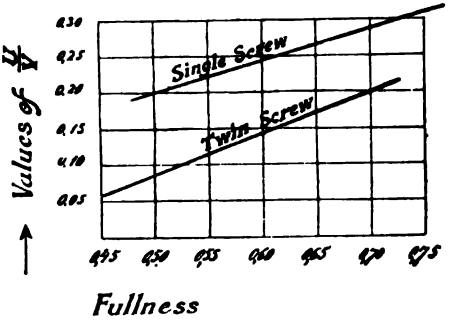


Fig. 347.

every revolution be retarded by a still greater amount than is shown in the formula $\frac{C-V}{C}$. The *real slip* is therefore—

$$s_t = \frac{C-(V-U)}{C}$$

in contradistinction to the *apparent slip*, which is always less, and is—

$$s_a = \frac{C-V}{C} = s_t - \frac{U}{C}$$

In ships with full stern lines the stream-line wake velocity U is large.

---

* Compare Taylor, "Resistance of Ships and Screw Propulsion."

† The formula given by Riehn (*Zeitschrift des Vereines Deutscher Ingénieure*, 1884, p. 469) cannot be used for the direct calculation of the "wake" as here defined, because he there attaches a somewhat different meaning to the term "wake" (*Vorstrom*).

If at the same time the real slip, as the result of a badly constructed screw (compare page 373), is very small, it may happen that there is no apparent slip, or that its value becomes negative. Real slip of course must always exist; it can never become zero or a negative quantity.

Example.—Suppose the speed of a ship, during its trial trip, to have been 10 knots, and the speed of the screw, calculated, from the pitch and the number of revolutions, to be

$$c = \frac{\pi \cdot H \cdot 60}{6{,}086{\cdot}44} = 9{\cdot}5 \text{ knots.}$$

The apparent slip is thus negative, and works out at

$$s_s = \frac{c - v}{c} = \frac{9{\cdot}5 - 10}{9{\cdot}5} = -5{\cdot}3\ ^{\circ}/_{\circ}.$$

The customary explanation is as follows:—The lines of the ship's stern are very full, there being but one screw, which is placed close to the stern-post. The effect of the wake current is therefore strongly marked. If the real slip be taken at only 10 $^{\circ}/_{\circ}$ (compare page 373), the wake current of the ship is—

$$u = c(s_t - s_s) = 9{\cdot}5(0{\cdot}1 + 0{\cdot}053) = 1{\cdot}45 \text{ knots.}$$

Negative slip is always a sign that the propulsion of the ship is inefficient, for it shows firstly that the real slip is small, which, according to § 218 (4), is unfavourable; secondly, such slip only occurs when the wake current is large. When such is the case, more energy is absorbed by the wake than is recovered from it.*

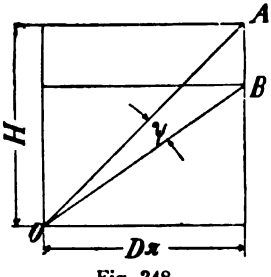


Fig. 348.

Angle of Slip (see Fig. 348).—During *one* revolution, corresponding to the path $D\pi$ of the tip of the blade, the screw moves forward in an axial direction over a distance H, if there is no slip; if there is slip, it moves only through the distance H – AB. The angle $AOB = \psi$ is called the slip angle.

§ 218. **Propeller Efficiency.**—If I.H.P. is the indicated horse-power of an engine, then B.H.P. $= \eta$. I.H.P. (see page 4) is the useful power expended in rotating the screw. A part only of this power is converted by the screw into axial or forward thrust,

* This customary explanation of "negative slip" must be used with caution, as it would be very difficult to demonstrate numerically that the wake can cause the slip to be less than 0.

the remainder is lost in overcoming frictional and other blade resistances. The efficiency of the *screw* is therefore—

$$\eta_s = \frac{\text{effective axial thrust of screw}}{\text{brake horse-power}} = \frac{\text{S.H.P.}}{\text{B.H.P.}}$$

For calculating the *efficiency** from the equations given for S.H.P. and B.H.P., see § 222.

In well-designed screws the efficiency varies from 40 to 70 °/₀; the lower values (40 to 50 °/₀) are for small screws in small ships, with engines running at a high speed; the higher values (55 to 70 °/₀) being for ships with large screws and running at moderately high speeds.†

The efficiency depends—

(1.) Upon the ratio between the pitch and the diameter, *i.e.*, upon the inclination of the elements of the blade to the plane perpendicular

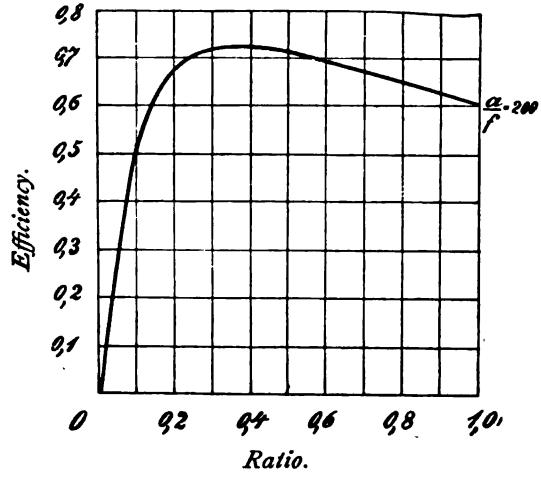


Fig. 349.

to the shaft. Fig. 349 (from Taylor) shows the efficiency of *elements of the blade* having varying ratios $\frac{d}{H}$.

The efficiency of those elements in which $\frac{d}{H}$ is less than about 0·15 is here seen to be very low, therefore the *extreme* ratio $\frac{D}{H}$ must be such that, for as large a number as possible of the blade elements, the

* For calculating the efficiency, see Riehn, *Zeit. des Ver. Deutscher Ing.*, 1884.

† A high speed within certain limits does not of itself involve a bad efficiency; but in those types of ships in which the engines run at high speeds, the conditions are unfavourable to the efficient working of the screw.

ratio $\frac{d}{H}$ conduces to a high efficiency, and lies between the limits $\frac{d}{H} = 0{\cdot}2$ to $0{\cdot}9$. In accordance with this rule, the extreme ratio of $\frac{D}{H}$ is generally found to be from $0{\cdot}5$ to $1{\cdot}2$.

Instances of how largely $\frac{D}{H}$ varies in practice, and between what wide limits it may lie, are shown in the following table, based upon results obtained from a large number of actual screws. Thus in

Torpedo-boats	having one screw	$\frac{D}{H} =$	$0{\cdot}8$ to $1{\cdot}1$.
Destroyers	„ two screws	„ =	
Small cruisers	„ two „	„ =	$0{\cdot}7$ „ $1{\cdot}0$.
Large cruisers and ironclads	„ two „	„ =	$0{\cdot}7$ „ $1{\cdot}1$.
Fast steamers	„ two „	„ =	$0{\cdot}6$ „ $0{\cdot}8$.
Freight and passenger steamers	„ two „	„ =	$0{\cdot}7$ „ $0{\cdot}95$.
Cargo boats	„ one screw	„ =	$0{\cdot}7$ „ $1{\cdot}0$.

In choosing the ratio $\frac{D}{H}$ the only thing to be considered is that the value $\frac{d}{H}$ is not too low for the greater part of the blade.

(2.) From the above remarks it follows that the efficiency also depends upon the shape of the *developed area* of the blade. If the greater part of this area lay close to the root of the blade, it would contain many elements in which the ratio $\frac{d}{H}$ would be unfavourable to the best efficiency. Therefore the blade must be small at the root. For constructive reasons we cannot go too far in this direction, neither must it be too broad at the tip, otherwise it might be easily broken. If, on the other hand, it is made too thick, the resistance of the leading edge prejudicially affects the efficiency. (Compare the formula for η_s § 222.) It is thus self-evident that the best blades are of the form most generally met with in practice, the developed area of which varies little from that of the normal or standard blade (Fig. 351). Small deviations from the normal shape of the developed area * only slightly affect the efficiency.

(3.) The ratio $\frac{\text{developed area}}{\text{area of disc}}$ or $k = \frac{zA}{\frac{\pi}{4}D^2}$ affects the efficiency.

* By "shape of the developed area," the shape of the boundary lines of the developed area is always to be understood.

If k is very large, it naturally follows that more of the area lies close to the boss than if, with the same total area of the blade, k were smaller; thus, if the value of k be high, the efficiency is low. If k be large, the blades must lie close together, and disturb each other by the eddies they create (compare the smaller values of a with many-bladed screws in the formulæ § 222).

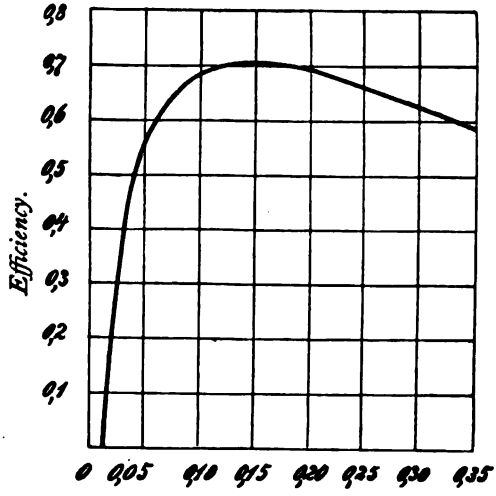


Fig. 350.

For the foregoing reasons the diameter of the screw is made as large as possible; the difficulty of doing this with the bulging full sterns of torpedo-boats should be noted. Only where it is impossible to increase the diameter,* and where the use of two screws is not desirable, do we find high values of k (or of b, see page 380).

The usual values of k are—

For torpedo-boats and small fast steamers	$k = 0{\cdot}30$ to $0{\cdot}33$
„ small warships	$k = 0{\cdot}33$ „ $0{\cdot}39$
„ ironclads	$k = 0{\cdot}32$ „ $0{\cdot}38$
„ fast steamers, mail steamers	$k = 0{\cdot}33$ „ $0{\cdot}39$
„ cargo steamers	$k = 0{\cdot}30$ „ $0{\cdot}45$
„ flat-bottomed river steamers and tugs	$k = 0{\cdot}40$ „ $0{\cdot}80$

Warships which are to be navigated in shallow water have, under certain circumstances, higher values of k than those given above.

(4.) The *real slip* s_1 also affects the efficiency, as may be deduced from the formula for efficiency η_s (see § 222). The most favourable *real slip is about* 15 to 20 %.

The connection between the efficiency and the slip for a blade-element having the ratio $\frac{d}{H} = 0{\cdot}6$ is shown in Fig. 350. (Compare Taylor, see note, page 369).

(5.) In ships with very small screws and running at very high

* As is the case in flat-bottomed river steamers and ferry boats. In these the blades are very wide, and their edges overlap and almost cover each other.

speeds (such as those driven by steam turbines), the efficiency may be greatly affected by what is known as "cavitation."* This term denotes the formation of hollow spaces or cavities on the forward side of the blades, due to the cutting by the propeller of the column of water set in motion.

§ 219. **Other Formulæ** for calculating the dimensions of ship's screws are generally based on the equation—

$$\text{I.H.P.} = c \cdot D^2(n \cdot H)^3.$$

The constant c, which varies for different types of ships, depends on the shape of the screw and of the ship (wake). The following formulæ are those generally used†—

$$D = K_1 \sqrt{\frac{\text{I.H.P.}}{\left(\frac{n \cdot H}{100}\right)^3}}, \text{ and } sA = K_2 \sqrt{\frac{\text{I.H.P.}}{n}}$$

Here D and H must be stated in feet, A in square feet.

The constants K_1 and K_2 are obtained from the following table:—

TABLE No. 40.

Coefficients K_1 *and* K_2 *for Calculating Screws.*

(From Seaton and Rounthwaite's "Pocket-Book of Marine Engineering.")

Type of Ship.	Speed of Ship v in Knots.	Number of		K_1.	K_2.	Material of Blades.
		Screws.	Blades.			
1. Cargo steamer with full lines	8 to 10	1	4	17 to 17·5	19 to 17·5	Cast iron.
2. Cargo steamer with moderately full lines	10 ,, 13	1	4	18 ,, 19	17 ,, 15·5	
3. Mail and passenger steamers, fine lines	13 ,, 17	1	4	19·5 ,, 20·5	15 ,, 13	Cast iron, bronze, or cast steel.
4. Do. do.	13 ,, 17	2	4	20·5 ,, 21·5	14·5 ,, 12·5	
5. Do., very fine lines	17 ,, 22	1	4	21 ,, 22	12·5 ,, 11	Gunmetal or bronze.
6. Do. do.	17 ,, 22	2	3	22 ,, 23	10·5 ,, 9	
7. Warships with very fine lines	16 ,, 22	2	4	21 ,, 22·5	11·5 ,, 10·5	
8. Do. do.	16 ,, 22	2	3	22 ,, 23·5	8·5 ,, 7	
9. Torpedo-boats, do.	20 ,, 26	1	3	25	7 ,, 6	Bronze or forged steel.

* For further details see Barnaby, "Marine Propellers."

† For similar formulæ see Fliege, *Zeitschrift des Vereines Deutscher Ingénieure*. 1893, p. 1552.

The values for fast steamers with two screws of four blades each come between lines 6 and 7 of Table No. 40.

Example.—Twin-screw fast steamer "Kaiser Wilhelm der Grosse." Let us assume the I.H.P. (calculated from the ship's resistance) = $2 \times 14{,}000$, and further $\frac{D}{H} = 0{\cdot}636$ (see § 218), and $n = 78$. Assuming that each screw has four blades, then $z = 4$. The diameter, pitch, and blade area are to be determined. From the above formula for D—

$$D^2 = K_1^2 \left(\frac{\text{I.H.P.}}{\frac{n \cdot H}{100}}\right)^3$$

Taking the value for $\frac{D}{H}$ as 0·636—

$$(0{\cdot}636)^2 \cdot H^2 = K_1^2 \frac{\text{I.H.P.}}{\left(\frac{n \cdot H}{100}\right)^3}$$

whence

$$H^5 = \frac{K_1^2 \cdot \text{I.H.P.} \cdot 100^3}{(0{\cdot}636)^2 \cdot n^3}$$

Solving the last equation, and taking the value of $K_1 = 22{\cdot}75$ (mean value from Table No. 40), H = about 32·8 feet.

In practice H = 33·5 feet (10·2 metres).

Further $D = 0{\cdot}636H = 20{\cdot}8$ feet (6·36 metres).

In practice D = 21·32 feet (6·5 metres).

The developed area is obtained from the equation $z_A = K_2\sqrt{\frac{\text{I.H.P.}}{n}}$.

According to Table No. 40, the mean of $K_2 = 10{\cdot}5$, whence

$$z_A = 10{\cdot}5\sqrt{\frac{14{,}000}{78}} = 141 \text{ square feet (13·4 square metres).}$$

In practice the total developed area of the blades is—

$$z_A = 136{\cdot}4 \text{ square feet (12·68 square metres).}$$

Calculation of the Area of the Screw from the pressure on the screw. The developed area of the blade may also be determined by taking as the basis of the calculation a pressure per square foot of blade area deduced from actual experiments. A large number of tests have yielded the following values for the ratio—

$$K_3 = \frac{\text{indicated thrust}}{\text{developed area of the screw}}$$

In torpedo-boats	it is = about	13	lb. per square inch.		
„ fast steamers	„ = „	10	„	„	
„ cargo and passenger steamers	„ = „	7 to 8·5	„	„	
„ cargo boats	„ = „	5·5 to 7	„	„	

§ 220. **Remarks.**—It is always better to make the pitch too small than too large. If the value H be smaller than the corresponding values given in the formula in § 219, those ratios which prejudicially affect the efficiency are avoided. On the other hand, by assuming a relatively smaller pitch, the only difficulty is, that the number of revolutions is increased beyond that assumed in the calculations, but the engines will then be better able to utilise to the full the power of the boilers. If H be larger than its calculated value, it may happen that the cylinders are too small to produce the turning moment necessary to give the required speed. Hence the engine will not be able to run at the required number of revolutions, and cannot impart the desired speed to the ship, although the boiler may easily be able to supply sufficient power for this speed.

§ 221. **Taylor's Method for Calculating a Ship's Screw.**—Although the formulæ given in § 219 are partly empirical, those given below for calculating the screw, taken from Taylor's book, are strictly theoretical. They confirm the empirical formulæ for I.H.P., from which they differ only because, instead of the constant K_1, expressions occur based on the diameter ratio, slip, coefficient of friction, &c. In Taylor's formulæ the use of indeterminate coefficients (such as those in which the physical significance is not clear) is unnecessary, as every coefficient can be deduced for each separate case from a physical basis. Taylor's method of calculation necessitates the use of experimental values for the stream-line wake (compare page 368); and as only constants having a definite meaning are used, this method, if followed, shows where the weak points of the calculation lie, and thus any great errors may be avoided.

§ 222. **Taylor's Theoretical Formulæ.**—Taylor* deduces the following formulæ theoretically:—

B.H.P. = effective work done in turning the screw

$$= 3 \times z\left(\frac{H \cdot n}{1{,}000}\right)^3 \times D_2(as_t X_c + fZ_c).$$

S.H.P. = effective or useful work due to the thrust of the screw (along the ship's course)

$$= 3 \times z\left(\frac{H \cdot n}{1{,}000}\right)^3 \times D_2\{as_t(1 - s_t)X_c - f(1 - s_t)Y_c\}.$$

* The following deductions are taken from the excellent work of D. W. Taylor, "Resistance of Ships and Screw Propulsion."

In both formulæ D and H must be stated in feet, B.H.P. and S.H.P. in horse-power, s_t as a decimal fraction; whilst z, n, a, f, X_c, Y_c, and Z_c are numerical values. The efficiency of the screw η_s is obtained by dividing one formula by the other—

$$\eta_s = (1 - s_t) \times \frac{a \cdot s_t \cdot X_c - f \cdot Y_c}{a \cdot s_t \cdot X_c + f \cdot Z_c}$$

If η is the efficiency of the engine (see page 4) then—

$$\text{B.H.P.} = \eta \times \text{I.H.P., therefore S.H.P.} = \eta \times \eta_s \times \text{I.H.P.}$$

In the above formulæ, the following symbols are used in addition to those given on page 361 :—

a = the so-called "coefficient of thrust" (thrust constant), and

$a = 8{\cdot}4 - 1{\cdot}0\frac{D}{H}$ in four-bladed screws.

$a = 9{\cdot}4 - 1{\cdot}2\frac{D}{H}$ in three-bladed screws.

$a = 10{\cdot}4 - 1{\cdot}4\frac{D}{H}$ in two-bladed screws

f is the coefficient of friction and of resistance due to the shape of the blade, the mean value of which may be taken at $f = 0{\cdot}045$.

X_c, Y_c, and Z_c are values which depend upon the ratio between the diameter and the pitch $\left(\frac{d}{H}\right)$ of the separate elements of the blade (see page 363) and on the shape and area of the blade. They may be expressed by the following equations—

$$X_c = \int X \frac{b}{d} \cdot \frac{dr}{d}\,; \quad Y_c = \int Y \frac{b}{d} \cdot \frac{dr}{d}\,; \quad Z_c = \int Z \frac{b}{d} \cdot \frac{dr}{d}\,.$$

Here b denotes the width of the developed blade, with radius $\frac{d}{2} = r$, and X, Y, and Z are functions of the ratio $\frac{d}{H}$ for this radius. The values for X, Y, and Z for various ratios $\frac{d}{H}$ are given in the following table :—

TABLE No. 41.—*Values of* X, Y, Z.

(See Taylor, " Resistance of Ships and Screw Propulsion.")

Diameter Ratio $\frac{d}{H}$	X.	Y.	Z.
0·1	0·077	1·048	0·10
0·2	0·288	1·181	0·47
0·3	0·582	1·374	1·22
0·4	0·912	1·606	2·54
0·5	1·254	1·862	4·60
0·6	1·598	2·134	7·58
0·7	1·939	2·416	11·68
0·8	2·277	2·705	17·09
0·9	2·612	2·999	23·98
1·0	2·944	3·297	32·54

TABLE No. 42.

Values of X_f, Y_f, Z_f *for the Standard Blade* (see Fig. 351).

$\frac{D}{H}$ Diameter Ratio.	0·00	0·01	0·02	0·03	0·04	0·05	0·06	0·07	0·08	0·09
					Values of X_f.					
0·4	0·170	0·176	0·183	0·189	0·195	0·202	0·208	0·215	0·222	0·229
0·5	0·236	0·243	0·250	0·257	0·264	0·272	0·279	0·286	0·294	0·301
0·6	0·309	0·316	0·323	0·331	0·338	0·345	0·353	0·361	0·368	0·376
0·7	0·383	0·391	0·398	0·406	0·414	0·422	0·430	0·438	0·445	0·453
0·8	0·461	0·469	0·471	0·479	0·492	0·500	0·508	0·516	0·523	0·531
0·9	0·539	0·546	0·555	0·562	0·570	0·578	0·586	0·593	0·601	0·609
					Values of Y_f.					
0·4	0·524	0·527	0·530	0·533	0·537	0·540	0·543	0·547	0·550	0·553
0·5	0·557	0·560	0·564	0·568	0·572	0·576	0·580	0·585	0·589	0·593
0·6	0·598	0·653	0·608	0·613	0·618	0·623	0·628	0·634	0·640	0·645
0·7	0·652	0·658	0·664	0·670	0·676	0·682	0·689	0·695	0·702	0·708
0·8	0·714	0·721	0·727	0 733	0·740	0·746	0·752	0·759	0·765	0·772
0·9	0·778	0·784	0·791	0·797	0·804	0·810	0·817	0·823	0·830	0·837
					Values of Z_f.					
0·4	0·48	0·50	0·51	0·52	0·54	0·55	0·57	0·59	0·61	0·63
0·5	0·65	0·68	0·70	0·73	0·75	0·78	0·81	0·84	0·87	0·91
0·6	0·94	0·98	1·01	1·05	1·10	1·14	1·19	1·24	1·29	1·34
0·7	1·40	1·45	1·51	1·57	1·63	1·69	1·76	1·83	1·90	1·97
0·8	2·04	2·11	2·19	2·27	2·34	2·43	2·51	2·60	2·68	2·77
0·9	2·87	2·97	3·06	3·15	3·25	3·36	3·46	3·57	3·68	3·78

For ordinary work the calculation of the values x_c, y_c, z_c ("characteristics" of the propeller blade) by means of the above formulæ and Table No. 41 is generally too complicated. In almost all cases that occur in practice, it is sufficient to take an ordinary blade, that is a blade with a *standard developed shape* (Fig. 351), as a starting-point, and to calculate from a table the values of x_c, y_c, z_c. In order,

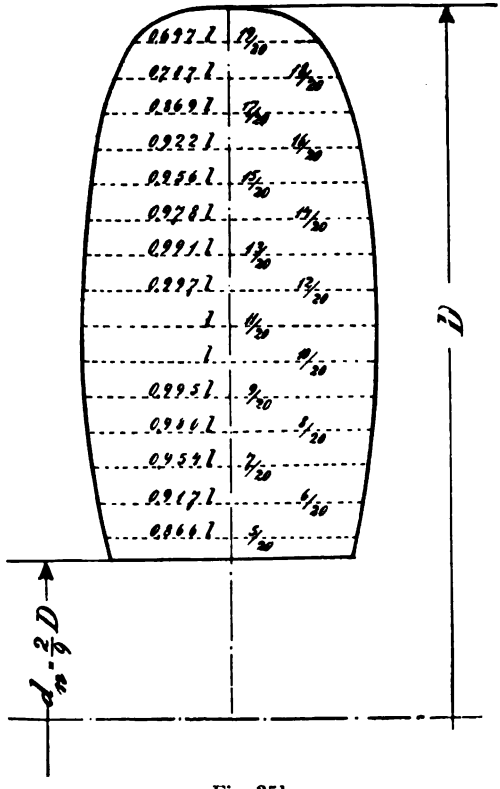


Fig. 351.

however, to make this standard type of blade independent of the area of the blade, we may write—

$$x_c = b x_f,\quad y_c = b y_f,\quad z_c = b z_f,$$

b being called the mean width ratio, a value which determines the ratio of width to length of the blade. (Compare page 363.)

The values x_f, y_f, z_f for the standard blade can be taken from Table No. 42. (The blades shown in Figs. 359 to 370 agree suffi-

ciently with general practice to form a basis for practical calculations.) It should be noted that the standard blade can only be used to determine the ratio of the width of each section of the blade to its maximum width. If b is known, x_c, y_c, and z_c can be at once calculated from Table No. 42.

By introducing the values x_f, y_f, z_f we get the following formulæ for calculating the dimensions of the screw and its efficiency—

$$\text{B.H.P.} = 3 \times z\left(\frac{\text{H} \cdot n}{1{,}000}\right)^3 \text{D}^2 b(as_t x_f + f z_f).$$

$$\text{S.H.P.} = 3 \times z\left(\frac{\text{H} \cdot n}{1{,}000}\right)^3 \text{D}^2 b[as_t(1 - s_t)x_f - f(1 - s_t)y_f].$$

For calculating the dimensions of the screw only the formula for B.H.P. is required.

Method of Calculation.—B.H.P. is calculated from the formulæ for the resistance of the ship. (See § 211.) By a similar method the wake is then determined. (See § 217.)

The speed due to the *screw* is therefore $\text{V} - \text{U}$. A real slip is next assumed. See § 218 (4). Then—

$$(1 - s_t)n\text{H} = \frac{(\text{V} - \text{U})6{,}086{\cdot}44}{60}$$

and from this the value of $n\text{H}$ can be obtained. A number of blades of a given shape are next assumed (the shape should conform as nearly as possible to that of the standard blade). This will determine x_f and z_f, f being always $= 0{\cdot}045$. As a rule the number of revolutions is also assumed. This gives the pitch—

$$\text{H} = \frac{(\text{V} - \text{U})6{,}086{\cdot}44}{n \cdot (1 - s_t) \cdot 60}$$

$\frac{\text{D}}{\text{H}}$ is so chosen, in accordance with the data given in § 218 (1), that a suitable diameter is obtained. Thus—

$$\text{D} = \frac{\text{D}}{\text{H}} \times \text{H}.$$

These are all the assumptions which may be made with safety. b can then be obtained from the formula for B.H.P., which determines the *area of the blade*. If b had been assumed, instead of the ratio $\frac{\text{D}}{\text{H}}$, the diameter D might have been determined from the formula for B.H.P. If both $\frac{\text{D}}{\text{H}}$ and b are assumed, the number of revolutions n can be calculated from the same formula.

The mean width ratio b may be selected from the table on page 373 for

$$k = \frac{z \cdot A}{\frac{\pi}{4} D^2}$$

According to the definition there given—

$$b = \frac{A}{D\left(\frac{D - d_n}{2}\right)}$$

Now as $d_n = cD$, and

$c = 0{\cdot}14$ to $0{\cdot}17$ for bosses with blades cast with them.
$c = 0{\cdot}24$ „ $0{\cdot}26$ „ „ screwed on.

Therefore

$$\frac{b}{k} = \frac{\frac{A}{D^2\left(\frac{1-c}{2}\right)}}{\frac{z \cdot A}{\frac{\pi}{4} D^2}} = \frac{\pi}{2z(1-c)}$$

With the help of this formula and the data given for k on page 373, b may be determined.

§ 223. **Example of Taylor's Method of Calculating the Dimensions and Shape of the Screw.**—Let this be done, for instance, for the S.S. "Deutschland." A speed of 23 knots is required. From the formulæ for ship's resistance, the indicated horse-power (I.H.P.) for this speed is given as 36,000. Thus for *one* engine, B.H.P. = $0{\cdot}9 \times 18{,}000 = 16{,}200$. From actual practice, with ships of similar build, a wake current of $\frac{U}{V} = 12$ °/₀ may be assumed for this class of vessel.

The speed, relative to the surrounding water, which the screw has to impart to the ship is thus—

$$V - U = 23(1 - 0{\cdot}12) = 20{\cdot}24 \text{ knots.}$$

A slip corresponding to an efficient screw must be selected (see page 373).

Assuming a somewhat high value of s_t say = 25 °/₀, then—

$$0{\cdot}75 n_H = 20{\cdot}24 \frac{6{,}086{\cdot}44}{60}$$

Whence $n_H = 2{,}733$ feet, and $c = \frac{n \cdot H \cdot 60}{6{,}086{\cdot}44} = 27$ knots.

This real slip of 25 °/₀, with a wake current of 12 °/₀, would correspond to an apparent slip of

$$s_e = s_t - \frac{U}{C} = 25\ ^{\circ}/_{\circ} - 10{\cdot}2\ ^{\circ}/_{\circ} = 14{\cdot}8\ ^{\circ}/_{\circ}.$$

This slip corresponds approximately to the apparent slip given by the results obtained on the trial trip.

The diameter ratio may now be assumed. In determining this, the shape of the stern post and the draught play an important part. Further, the curve giving the best diameter ratio must be taken into account (see page 371). Allowing for all these, and having regard also to circumstances limiting the number of revolutions, a diameter ratio of $\frac{D}{H} = 0{\cdot}65$ may be chosen.

For further calculations we return to the original formula—

$$\text{B.H.P.} = 3 \times z\left(\frac{n \cdot H}{1{,}000}\right)^3 \cdot D^2 b(as_t x_f + fz_f).$$

Having regard to the solidity and safe working of the screw, we will take the number of blades as $z = 4$. The coefficient of thrust ("thrust constant") is thus—

$$a = 8{\cdot}4 - 0{\cdot}65 = 7{\cdot}75\ ;\ \text{and also}\ f = 0{\cdot}045\ \text{(see page 377)}.$$

The shape of the blade is assumed to be of the usual pattern (similar to the standard type of blade shown on page 379). Therefore the values x_f, y_f, z_f may be taken direct from Table No. 42, page 378, and with

$$\frac{D}{H} = 0{\cdot}65,\ \text{we get}\ x_f = 0{\cdot}345,\ y_f = 0{\cdot}623,\ z_f = 1{\cdot}14.$$

The number of revolutions n may be calculated from the formula for B.H.P., and the mean width ratio b assumed, or if the number of revolutions be assumed, a definite value will then be obtained for b, which determines the *area of the blade*. Assuming $n = 79{\cdot}5$, the pitch must now be calculated. As $n = 79{\cdot}5$, $n \times H = 2733$ feet, then H will be $= 34{\cdot}35$ feet (pitch of actual screw $H = 34{\cdot}44$ feet). Hence the diameter is obtained—

$$D = \frac{D}{H} \cdot H = 0{\cdot}65 \times 34{\cdot}35 = 22{\cdot}32 \text{ feet}$$

which corresponds exactly to the diameter of the screw as fitted in the ship. The mean width ratio b has next to be determined, and is obtained by working out the equation for B.H.P. and b.

The values within brackets become

$$(as_t x_f + fz_f) = (7{\cdot}75 \times 0{\cdot}25 \times 0{\cdot}345) + (0{\cdot}045 \times 1{\cdot}14) = 0{\cdot}719.$$

Therefore $$b = \frac{16{,}200}{4 \times 3 \times 0{\cdot}204 \times 498{\cdot}1 \times 0{\cdot}719} = 0{\cdot}185.$$

This value almost exactly corresponds to actual practice, in which the developed area of a blade is in fact 35 square feet. Its length from boss to tip $\frac{D - d_n}{2} = 8{\cdot}28$ feet. The mean width is $l_m = 4{\cdot}22$ feet, and thus the mean width ratio works out at

$$b = \frac{l_m}{D} = 0{\cdot}189.$$

The efficiency of this screw can now be calculated. It is as follows (see page 377)—

$$\eta_e = \frac{\text{S.H.P.}}{\text{B.H.P.}} = \frac{as_t(1 - s_t)x_c - f(1 - s_t)y_c}{as_t x_c + fz_c}$$

In this formula replace x_c by x_f and b will then be eliminated—

$$(x_c = bx_f, \text{ \&c.}).$$

Hence—

$$as_t(1 - s_t)x_f - f(1 - s_t)y_f = 7{\cdot}75 \times 0{\cdot}25 \times 0{\cdot}75 \times 0{\cdot}345 - 0{\cdot}045 \times 0{\cdot}75 \times 0{\cdot}623 = 0{\cdot}48.$$

The denominator of the fraction has already been determined above; the efficiency is therefore—

$$\eta_e = \frac{0{\cdot}48}{0{\cdot}719} = 0{\cdot}67.$$

Strength of Propeller Blades.

§ 224. **Stress in the Propeller Blade due to Thrust and Tangential Forces.**—The greatest stress is at the root of the blade, and for this reason it is always worked out for a point as near the boss as possible.

Two forces, a turning force at right angles to the direction of the shaft (tangential force T), and a force parallel to the shaft (thrust P), act upon each element of the blade. These forces can be summed up as a resultant thrust acting on a given point, which will produce the same bending moment on the root of the blade, as the sum of all the thrust forces on all the elements of the blade. The distance of this point from the centre of the shaft is—

$$k_1 \frac{D}{2} = \frac{\text{total thrust moment}}{\text{total thrust}}$$

In the same way the distance of the point from the centre at which the sum of all the tangential forces is supposed to be concentrated will be—

$$k_2 \frac{D}{2} = \frac{\text{total moment of tangential forces}}{\text{total tangential force}}$$

The constants k_1 and k_2 are found for the standard propeller blade from Table No. 43. (For this and the following passages compare Taylor.)

TABLE No. 43.

Constants.

$\frac{D}{H}$	k_1.	k_2.	$\frac{D}{H}$	k_1.	k_2.
0·4	0·706	0·646	0·8	0·688	0·614
0·5	0·710	0·658	0·9	0·695	0·606
0·6	0·692	0·644	1·0	0·696	0·600
0·7	0·684	0·625			

The *thrust* P is obtained from the following equation:—

Work due to the thrust per minute = S.H.P. = about 0·7 B.H.P.; for greater safety the maximum efficiency of 70 °/₀ is here taken. Therefore—

$$\frac{P \times H(1 - s_t) \times n}{33{,}000} = 0{\cdot}7 \text{ B.H.P.} \times \frac{1}{z} \text{ ft. lb. per minute.}$$

Taking $s_t = 20\ ^{\circ}/_{\circ} = 0{\cdot}2$ for the normal slip s_t, then—

$$P = \frac{28{,}875 \text{ . B.H.P.}}{z \times H \times n}$$

H being stated in feet, P in pounds, B.H.P. in horse-power.

The tangential force T for *one* blade is obtained from the well-known equation—

$$\text{Moment of } T = \frac{\text{B.H.P.}}{n} \times 5{,}252 \times \frac{1}{z}$$

Therefore

$$T \times k_2 \times \frac{D}{2} = \frac{\text{B.H.P.}}{n} \times 5{,}252 \times \frac{1}{z}$$

whence—

$$T = \frac{\text{B.H.P.}}{n} \times \frac{10{,}504}{k_2 \times D \times z}$$

D being in feet, T in pounds, and B.H.P. in brake horse-power.

If the section of the blade in question is at the distance $\frac{d}{2}$ from the axis, then the—

$$\text{Bending moment due to } P = P\left(k_1 \frac{D}{2} - \frac{d}{2}\right)$$

$$\text{,,} \qquad \text{,,} \qquad T = T\left(k_2 \frac{D}{2} - \frac{d}{2}\right)$$

To calculate the dimensions of the cross section of the blade, it is necessary to know the bending moments about the longitudinal axis XX, and about the transverse axis YY (Fig. 352). These are—

$$\text{Bending moment about XX, } M_1 = M_T \sin \alpha + M_P \cos \alpha$$

$$\text{,,} \qquad \text{,,} \qquad \text{YY, } M_2 = M_P \sin \alpha - M_T \cos \alpha.$$

The curve at the root of the blade is generally taken as a parabola having an area $= \frac{2}{3} lh$ (Fig. 353).

Distance of the centre of gravity from AB, $d = \frac{2}{5} h$. Moment of inertia about the axis through the centre of gravity, and parallel to AB $= \frac{8}{175} lh^3$. Moment of inertia about the axis CD $= \frac{1}{30} l^3 h$.

From these values we get—

(*a.*) Moment of resistance for axis parallel to AB—

$$\text{For AB, } w_z = \frac{4}{35} lh^2 \text{; for C, } w_d = \frac{8}{105} lh^2$$

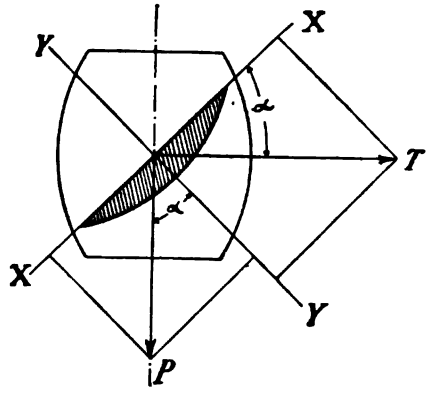


Fig. 352.

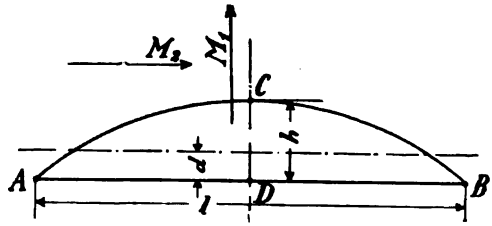


Fig. 353.

(*b.*) Moment of resistance for axis parallel to CD—

$$\text{For A and B} = \frac{1}{15} l^2 h.$$

Thence

$$\text{Tension at A and B due to } M_1 = \frac{35}{4} \frac{M_1}{lh^2}$$

$$\text{Compression at C due to } M_1 = \frac{105}{8} \cdot \frac{M_1}{lh^2}$$

$$\text{Tension at A due to } M_2 = \frac{15 M_2}{l^2 h}$$

$$\text{Compression at B due to } M_2 = \frac{15 M_2}{l^2 h}$$

The maximum tension at $A = s_{z_1} = \frac{35}{4} \cdot \frac{M_1}{lh^2} + \frac{15M_2}{l^2h}$

The maximum compression at $C = s_{d_1} = \frac{105}{8} \frac{M_1}{lh^2}$

Assuming a value for l, h can be calculated from the formula for s_z and s_{d_1}, the lower values of each being taken as a basis. The following table gives the allowable stresses for different materials :—

TABLE No. 44.

Allowable Stresses in Propeller Blades, due to Thrust and Turning Moment.

Material of the Blades.	Working Stress in pounds per sq. in.	
	s_{z_1} Tension.	s_{d_1} Compression.
Cast iron - - - - -	2,000	6,000
Cast steel - - - - -	5,000	10,000
Bronze - - - - -	3,000	4,000
Manganese or phosphor bronze -	5,000 to 8,500	6,000 to 12,000

The higher values given on the last line are only allowable in light warships, in which, on account of the weight being so much reduced, such high stresses are unavoidable.

§ 225. **For Working Calculations**, the above formulæ may be thus simplified :—

If $\frac{D}{H}$ be taken as $= 0{\cdot}6$, $k_2 = 0{\cdot}644$, then $T = 0{\cdot}94P$. To calculate the dimensions of the cross section of the blade, for which $a =$ about $45°$ (which generally holds for the root of the blade); assuming T and P to be exerted respectively at the distances $k_2\frac{D}{2}$ and $k_1\frac{D}{2}$ from the centre of the shaft, then—

$$M = \left(0{\cdot}66\frac{D}{2} - \frac{d}{2}\right) 1{\cdot}94P \cos 45°$$

$$= \left(0{\cdot}66\frac{D}{2} - \frac{d}{2}\right) 1{\cdot}4P$$

$$= \left(0{\cdot}66\frac{D}{2} - \frac{d}{2}\right) \frac{40{,}425 \times \text{B.H.P.}}{s \cdot H \cdot n}$$

where D, d, and H are in feet, B.H.P. in brake horse-power, and M in ft. lb.

If the ratio of $\frac{D}{H}$ is greater than 0·6, the value M must be multiplied by a factor less than unity. In such cases, instead of M we should have—

$$\begin{aligned} 0{\cdot}97\text{M} \text{ if } \frac{D}{H} &= 0{\cdot}65 \\ 0{\cdot}94\text{M} \quad \text{,,} &= 0{\cdot}7 \\ 0{\cdot}89\text{M} \quad \text{,,} &= 0{\cdot}8 \\ 0{\cdot}85\text{M} \quad \text{,,} &= 0{\cdot}9 \\ 0{\cdot}82\text{M} \quad \text{,,} &= 1{\cdot}0 \\ 0{\cdot}80\text{M} \quad \text{,,} &= 1{\cdot}1 \end{aligned}$$

The stress in the given cross section of the blade will thus be—

$$s_{z1} = \frac{35}{4} \times \frac{M}{lh^2}; \text{ and } s_{d1} = \frac{105}{8} \times \frac{M}{lh^2}$$

The bending moment about the transverse axis may be neglected. (Compare Example II., page 391.)

Example.—Twin-screw fast steamer "Deutschland." What is the maximum stress in the blade at a distance of say 4·1 feet (50 inches) from the centre of the shaft?

Let $D = 22{\cdot}3$ feet, $H = 34{\cdot}4$ feet, $n = 79{\cdot}5$, $\frac{d}{2} = 4{\cdot}1$ feet, $l = 4{\cdot}25$ feet,

$h = 0{\cdot}86$ feet, B.H.P. $= 16{,}200$, $s = 4$, and $\frac{D}{H} = 0{\cdot}65$.

$$\text{Therefore } M = 0{\cdot}97(0{\cdot}66 \times 11{\cdot}15 - 4{\cdot}1)\frac{40{,}425 \times 16{,}200}{4 \times 34{\cdot}4 \times 79{\cdot}5}$$

$$= 0{\cdot}97 \times \frac{3{\cdot}26 \times 40{,}425 \times 16{,}200}{4 \times 34{\cdot}4 \times 79{\cdot}5} = \text{about } 191{,}000 \text{ ft. lb.}$$

Hence the—

Maximum tension $s_{z1} = \frac{35}{4} \times \frac{191{,}000}{4{\cdot}25 \times {\cdot}74} = 533{,}000$ lb. per sq. ft.
$= 3{,}700$ lb. per sq. in.

Maximum compression $s_{d1} = \frac{105}{8} \times \frac{191{,}000}{4{\cdot}25 \times {\cdot}74} = 800{,}000$ lb. per sq. ft.
$= 5{,}560$ lb. per sq. in.
$= \text{about } 1{\cdot}5 s_{z1}$.

The material used is manganese bronze.

§ 226. **Stresses in the Blades due to Centrifugal Force.**—To the stresses s_{z1} and s_{d1} must be added the stress, due to the centrifugal force, produced by that part of the blade which lies outside the section now under consideration. Let G be the weight of this part (the shaded

area in Figs. 354 and 355), s its centre of gravity, w its circumferential speed, r the radius of the circle it describes. Then the centrifugal force acting on section xx (Fig. 354) is—

$$C = \frac{w^2}{r} \times \frac{G}{g}$$

This produces in the blade the tensile stress $s_{b_2} = \frac{C}{F}$ lb. per square inch ; F being the area of the cross section in square inches. The

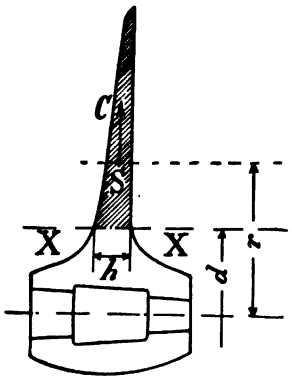


Fig. 354.

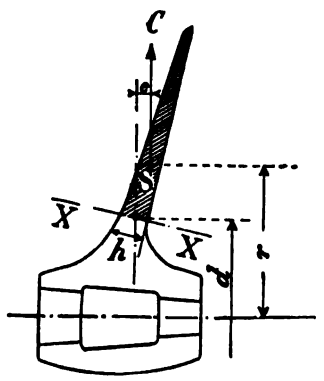


Fig. 355.

maximum tensile stress in the blade (Fig. 354) at section xx is at the points A and B (Fig. 353), and is—

$$s_z = s_{z1} + s_{z2}.$$

The maximum compression is at c and is—

$$s_d = s_{d1} - s_{z2}.$$

If the blade is inclined to the rear (Fig. 355), the centrifugal force also produces a bending moment in the longitudinal plane of the blade. The amount of this force on the section at the root is about $M_c = 0{\cdot}7ec$. If we take the section as forming a segment of a parabola (Fig. 353), then from this moment we get a tension at points A and B of—

$$s_{z3} = \frac{M_c}{W_z} = \frac{M_c}{\frac{4}{35} lh^2}$$

and a compression at c of—

$$s_{d3} = \frac{M_c}{W_d} = \frac{M_c}{\frac{8}{105} lh^2} = 1{\cdot}5 s_{z3}.$$

In a blade placed at such an angle, the greatest total tension at A and B, using the shortened formula, will be (for exact calculations the tension at A should be more than that at B, see page 386, § 224)—

$$\text{Maximum total tension} \qquad s_z = s_{z1} + s_{z2} + s_{z3}.$$
$$\text{Maximum total compression} \qquad s_d = s_{d1} - s_{z2} + s_{z3}.$$

The centre of gravity of that part of the blade lying outside the section here considered may approximately be taken as being at the distance $r = \frac{d}{2} + \left(\frac{D}{2} - \frac{d}{2}\right) 0{\cdot}3$ from the axis.

Those sections of the blade which are calculated only for bending stresses from the thrust and tangential force, must be tested, to see whether they are able to withstand stresses due to centrifugal force, and if necessary strengthened. For this reason the working stresses in Table No. 44 will be exceeded by 20 to 30 per cent.

§ 227. **Example I. Effect of Centrifugal Force** on the stress in the blades of the screw of the "Deutschland." The stress in a section of the blade is taken, as before, at a distance of 50 inches from the centre of the shaft.

Weight of blade outside this section is $G = 4{,}180$ lb.

Distance of the centre of gravity of this part from the shaft is $r = 6{\cdot}23$ feet, and the area of the section is $F = 357{\cdot}5$ square inches. If $n = 79{\cdot}5$, the circumferential speed will be—

$$w = 51{\cdot}8 \text{ feet per second.}$$

Therefore $$C = \frac{G}{g} \times \frac{w^2}{r} = 56{,}000 \text{ lb.}$$

Hence $$s_{z2} = \frac{C}{F} = \frac{56{,}000}{357{\cdot}5} = \text{about } 156 \text{ lb. per square inch.}$$

The blade is inclined to the rear, as shown in Fig. 355; the centrifugal force therefore exerts not only the tension C, but also a bending moment upon the section under consideration. $e = 11$ inches; the moment of resistance of the section about its axis parallel to AB (Fig. 353) is—

$$\text{Tension (in AB)} = \frac{4}{35} lh^2 = \text{about } 645 \text{ inches} = w_b = 1{\cdot}5 w_d.$$

$$\text{Compression (in C)} = \frac{8}{105} lh^2 = \text{about } 430 \text{ inches} = w_d.$$

Therefore $$s_{z3} = \frac{M_c}{w_z} = \frac{0{\cdot}7 \times e \times c}{w_z} = \frac{0{\cdot}7 \times 11 \times 56{,}000}{645}$$
$$= \text{about } 670 \text{ lb. per square inch.}$$

and $$s_{d3} = \frac{M_c}{w_d} = \frac{0{\cdot}7 \times e \times c}{w_d} = \frac{0{\cdot}7 \times 11 \times 56{,}000}{430}$$
$$= \text{about } 1{,}000 \text{ lb. per square inch.}$$

The total stresses upon the section are therefore—

Tension (in AB) $s_z = s_{z1} + s_{z2} + s_{z3} = 3{,}700 + 156 + 670$
$= 4{,}526$ lb. per square inch.

Compression (at c) $s_d = s_{d1} - s_{z2} + s_{d3} = 5560 - 156 + 1{,}000$
$= 6{,}404$ lb. per square inch.

It is thus shown that the stress in the blade is greatly increased by the centrifugal force, and this justifies the high factor of safety used in the figures given in Table No. 44.

§ 228. **Example II.**—The centrifugal force has much more effect on the blades of screws running at a high speed. In a recently constructed destroyer, in which $n = 325$; I.H.P. $= 2{,}760$; H $= 7{\cdot}85$ feet $= 94{\cdot}4$ inches; $l = 1{\cdot}64$ feet $= 19{\cdot}6$ inches; $r = 2{\cdot}18$ feet $= 26{\cdot}18$ inches; $e = 0{\cdot}26$ feet $= 3{\cdot}14$ inches; $\frac{D}{H} = 1{\cdot}08 =$ say $1{\cdot}1$; $z = 3$; D $= 8{\cdot}54$ feet $= 102{\cdot}3$ inches; $h = 0{\cdot}295$ feet $= 3{\cdot}54$ inches; $d = 2{\cdot}3$ feet $= 27{\cdot}54$ inches.

$\left.\begin{matrix} k_1 = 0{\cdot}7 \\ k_2 = 0{\cdot}6 \end{matrix}\right\}$ assumed values. (Compare Table No. 43.)

Hence B.H.P. $=$ I.H.P. $\times\ 0{\cdot}85 = 2{,}345$.

According to the formula in § 224—

$$T = \frac{\text{B.H.P.} \times 10{,}504}{n \cdot k_2 \cdot D \cdot z} = 4{,}930 \text{ lb.}$$

Bending moment due to $T = T\left(k_2 \frac{D}{2} - \frac{d}{2}\right) = 6{,}950$ ft. lb.

$$P = \frac{\text{B.H.P.} \times 28{,}875}{z \cdot H \cdot n} = 8{,}800 \text{ lb.}$$

Bending moment due to $P = P\left(k_1 \frac{D}{2} - \frac{d}{2}\right) = 16{,}200$ ft. lb.

From the graphic analysis, Fig. 356, we get—

Moment about the axis XX $= M_1 = 15{,}900$ ft. lb.
" " YY $= M_2 = 7{,}230$ "

These moments produce the following stresses—

Tension in AB due to $M_1 = \frac{35}{4} \times \frac{M_1}{lh^2} \times \frac{1}{144} = 6{,}750$ lb. per sq. in.

Compression at C „ $M_1 = \frac{105}{8} \times \frac{M_1}{lh^2} \times \frac{1}{144} = 10{,}200$ „ „

Tension at A „ $M_2 = 15 \times \frac{M_2}{l^2h} \times \frac{1}{144} = 950$ „ „

Compression at B „ $M_2 = 15 \times \frac{M_2}{l^2h} \times \frac{1}{144} = 950$ „ „

Hence the greatest tension at A $= s_{t1} = 6{,}750 + 950$
$= 7{,}700$ lb. per square inch.

„ „ compression at C $= s_{d1} = 10{,}200$ „ „

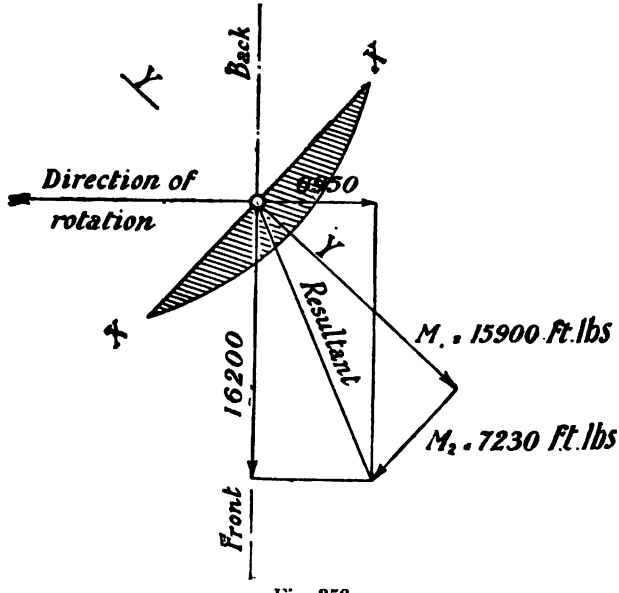


Fig. 356.

The weight of the part of the blade outside the section, at a distance $\frac{d}{2} = 1{\cdot}15$ feet from the boss $=$ G $= 352$ lb. Circumferential speed of the centre of gravity s of the part of the blade having weight G—

$$w = \frac{2\pi rn}{60} = \frac{2\pi \times 2{\cdot}18 \times 325}{60} = 74 \text{ feet per second.}$$

Therefore $$C = \frac{G}{g} \times \frac{w^2}{r} = \frac{352 \times 74^2}{32{\cdot}2 \times 2{\cdot}18} = 27{,}500 \text{ lb.}$$

Hence $$s_{z2} = \frac{C}{F} = \frac{27{,}500}{\cdot 325} \times \frac{1}{144} = 590 \text{ lb. per square inch nearly.}$$

(F = area of section $= \frac{2}{3} lh = \frac{1{\cdot}64 \times {\cdot}295 \times 2}{3} = {\cdot}325$ square foot.)

The moment of resistance of the section is—

For tension (in AB) $$w_z = \frac{4}{35} lh^2 = 28 \text{ inches}^3.$$

For compression (at c) $$w_d = \frac{8}{105} lh^2 = 18{\cdot}6 \quad ,,$$

The moment of the centrifugal force is very considerable here, because of the slanting position of the elements of the blade, and the great centrifugal force exerted ($e = 3{\cdot}14$ inches)—

$$s_{z3} = \frac{M_c}{w_z} = \frac{0{\cdot}7 \times 3{\cdot}14 \times 27{,}500}{28} = 2{,}160 \text{ lb. per square inch.}*$$

$$s_{d3} = \frac{M_c}{w_d} = \frac{0{\cdot}7 \times 3{\cdot}14 \times 27{,}500}{18{\cdot}6} = 3{,}250 \quad ,, \quad ,,$$

The total stress on this section of the blade is thus—

$$s_z = s_{z1} + s_{z2} + s_{z3} = 7{,}705 + 590 + 2{,}160 = 10{,}450 \text{ lb. per sq. inch.}$$
$$s_d = s_{d1} - s_{z2} + s_{d3} = 10{,}200 - 590 + 3{,}250 = 12{,}860 \quad ,, \quad ,,$$

The tensile stress due to centrifugal force is thus shown to be considerable in this case.

§ 229. **Thickness of Tip of Blade.**—For solid bronze this thickness h_o is (Fig. 357)—

$h_o = \frac{5}{16}$ to $\frac{1}{2}$ inch if D = from 6 to 10 feet.
$h_o = \frac{1}{2}$ to $\frac{3}{4}$,, if D = ,, 10 to 12 ,,
$h_o = \frac{3}{4}$ to $\frac{7}{8}$,, if D = ,, 12 to 19 ,,
$h_o =$ 1 ,, if D = > 19 ,,

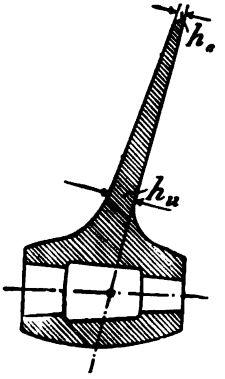


Fig. 357.

For cast iron the thickness of blade at the tip is about *one and a half times* the above. The crowns of the sections at the tip and root are joined by a straight line, and the thicknesses h for each section are thus determined. The back of each section is formed by the segment of a circle; it is also usual to round away the edges of the sections nearest the boss, on the working or *thrust face*. (See Figs. 361, 363.)

* This value, 2,160 lb. per square inch, is slightly raised by the components of the centrifugal force exerted along XX, but the increase is so small that it may be neglected.

The lowest part of the blade is rounded off into the flange with a very ample curve. The edges of the blade are made as sharp as possible, and in warships the blades are often finished bright all over.

§ 230. **Materials used for Blades.**—For all warships, and in general for first-class steamers, manganese or phosphor bronze is used. For cargo and passenger vessels of medium size, ice-breakers and large tugs, cast steel is employed; and cast iron for ordinary cargo boats, and for small and medium-sized lower class vessels. If cast iron be used, the boss and blade are cast in one. Cast steel has the disadvantage that it rusts easily, and also that it is liable to shrink during casting, and cause inaccuracies in the surface of the screw. In small steam tugs the blades are often made of steel plates, and riveted to the cast steel boss.

Construction of the Screw.

§ 231. **Moulding and Casting the Screw** (Fig. 358).—A vertical spindle, such as is used for loam moulding, is set up on a cast-iron base plate. On this spindle a striking board or arm is fitted in such a manner that it is free to rotate, and to slide up and down. This board, the striking edge of which sweeps out the surfaces of the blades, is placed with the edge at right angles or inclined to the spindle, according to the design of the screw. Round the spindle are fixed concen-

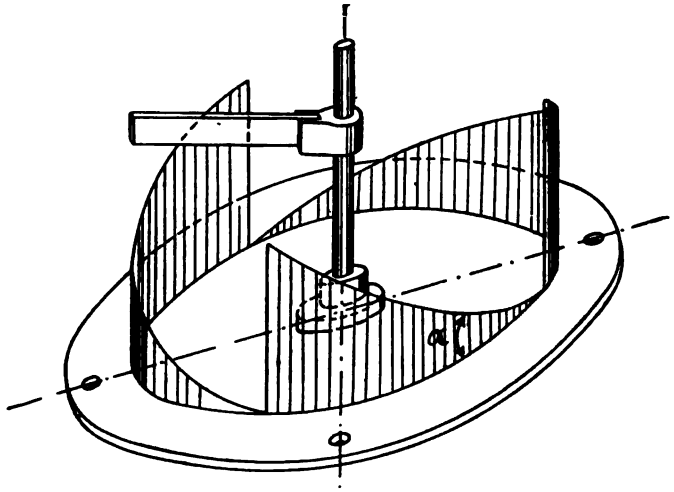


Fig. 358.

trically as many special templates as there are blades; each template being a triangular metal blade, bent to a radius somewhat larger than $\frac{D}{2}$. The angle of inclination α of their upper edges gives the required pitch of the screw. The space between the spindle and the ring of templates is filled in with loam. The striking board is then turned, sliding on the templates which remain fixed, and as it revolves its combined motion causes the loam to take the required helical

form for the face of the screw blades. The pattern thus obtained for the back or thrust surfaces of the blades is baked; the centre lines of the blade, and circles concentric to the striking spindle, are then drawn on its surface. Upon these circles, and symmetrically about the centre lines of the blades, thin metal templates are erected at right angles to the moulded surface, representing the cross section of the blade at the various distances from the centre. For the shape of the latter see Figs. 361, 363. The spaces between these are filled with loam, and the forward surface of the screw blades thus obtained is smoothed off, dried, and painted, and used as the pattern for the upper part of the mould. When this part is complete, the sectional templates and dried loam are removed from the lower surface, and a finished mould of the blade remains, which only requires the addition of the boss or the flange to complete it.

§ 232. Explanation of the Drawings of Screws.

1. Figs. 359 to 362 show the screw of a small tug with *one* engine; the material used for the screw being cast steel.

$$\text{I.H.P.} = 200;\ v = 7 \text{ knots};\ n = 125;\ D = 6{\cdot}56 \text{ feet};$$
$$H = 8{\cdot}53 \text{ feet};\ z = 4;\ z_A = 21{\cdot}5 \text{ square feet.}$$

The area of the blade in Fig. 359 is developed according to Method I., § 214. The blades are slightly inclined to the rear. (See Fig. 362.)

2. Figs. 363 to 366 show the screw of a large twin-screw mail steamer (passenger and cargo boat). The blades are of manganese bronze; the boss of cast iron.

$$\text{I.H.P.} = 2 \times 4{,}500;\ v = 14{\cdot}5 \text{ knots};\ n = 80;\ D = 18{\cdot}7 \text{ feet};$$
$$H = 21{\cdot}3 \text{ feet};\ z = 4;\ z_A = 99 \text{ square feet.}$$

The area of the blade is developed according to Method I., § 214. The studs for securing the blade are of Siemens-Martin steel; the corresponding cap nuts of forged bronze. The boss is strengthened at the front and back by strong wrought-iron rings shrunk on. The holes in the flange of the blade are oval, so that the pitch of the screw can be varied from 20 to 22·7 feet. Here, again, the blades are slightly inclined to the rear.

3. Figs. 367 to 370 show the screw of a twin-screw armoured cruiser. The blades and boss are of high tension bronze.

$$\text{I.H.P.} = 2 \times 8{,}000;\ n = 140;\ v = 21 \text{ knots};\ z = 3;$$
$$D = 17{\cdot}4 \text{ feet};\ H = 16{\cdot}8 \text{ feet.}$$

The blades are vertical to the axis of the screw.

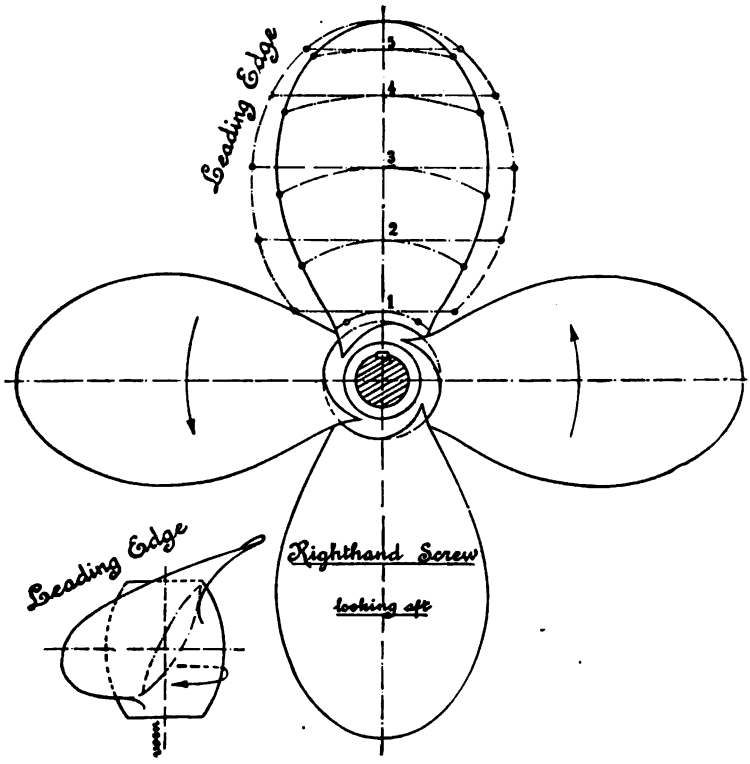


Fig. 359.

Fig. 360.

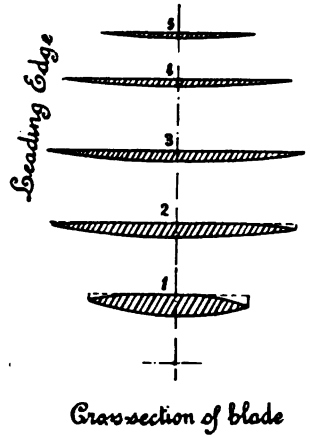


Fig. 361.

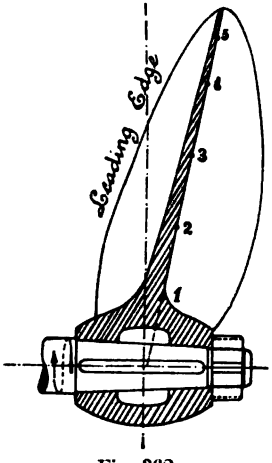


Fig. 362.

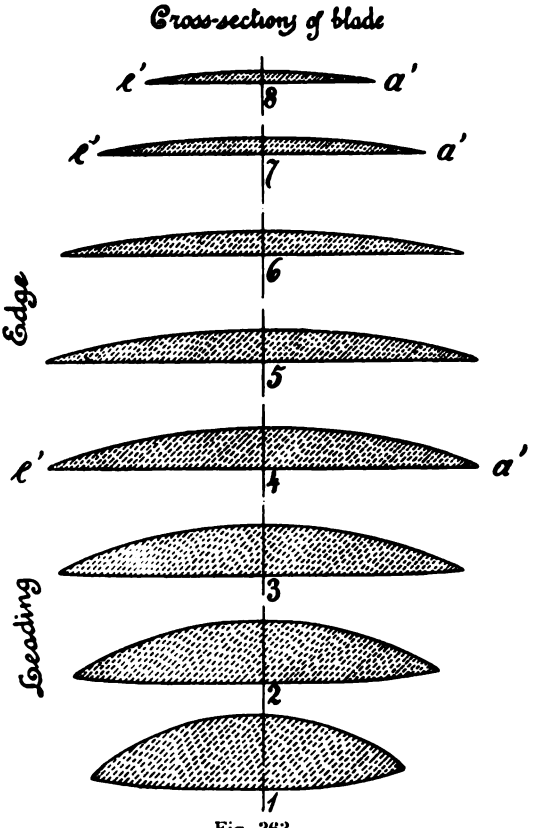


Fig. 363.

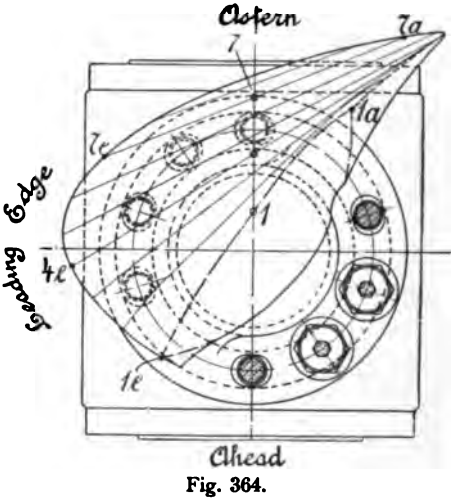


Fig. 364.

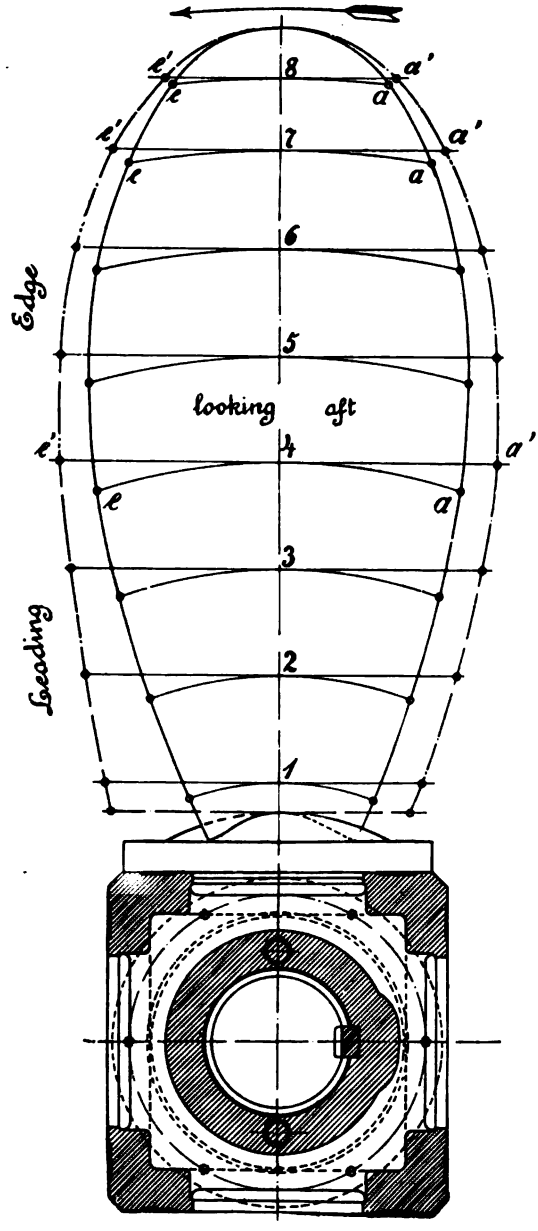


Fig. 365.

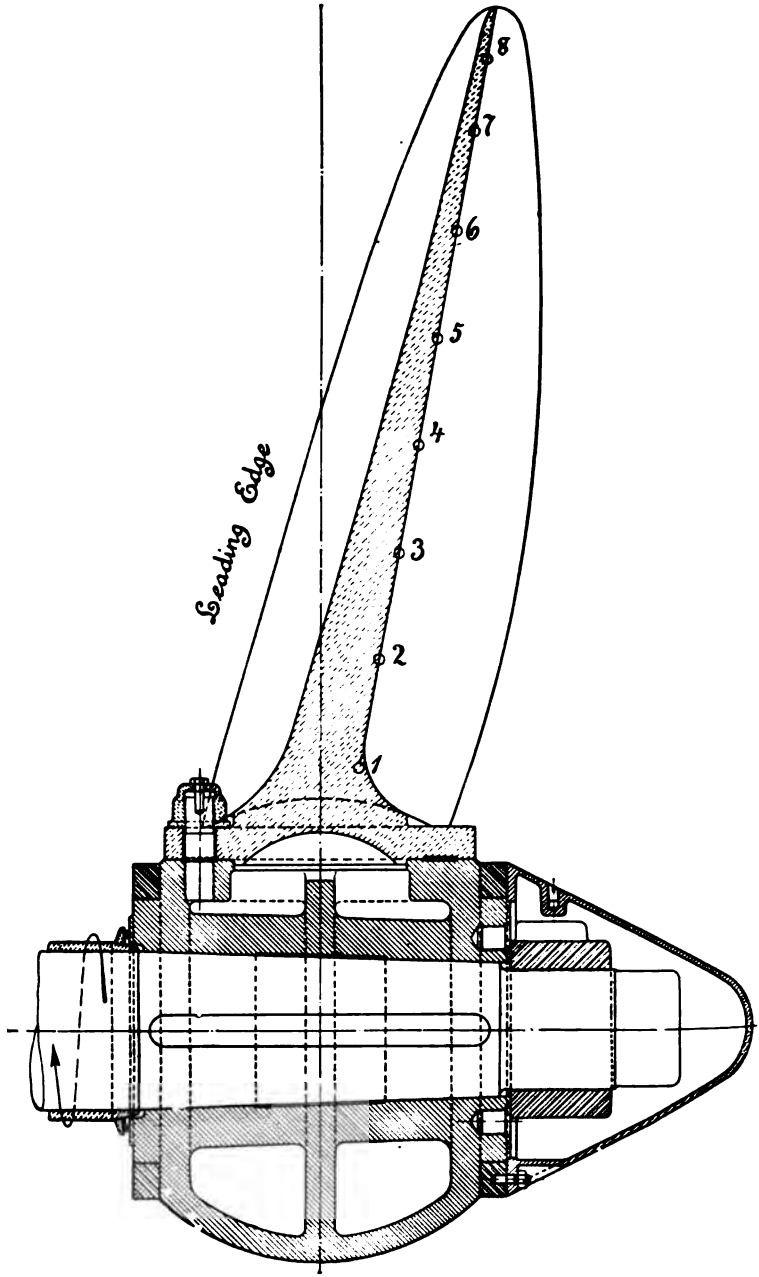


Fig. 366.

§ 233. **Propeller Boss.**—1. In smaller propellers the boss and blades are cast in one (see Fig. 362). Length of boss $L = 2{\cdot}3$ to $2{\cdot}6d$. Maximum diameter of boss $d_n = 2{\cdot}0$ to $2{\cdot}3d$. Slope of cone of the propeller shaft 1 in 10 to 1 in 16.

As a rule the centre part of the boss is cut away, firstly to effect a saving of weight, secondly to facilitate the fitting of the propeller on to the conical end of the shaft. In order that the turning moment of the

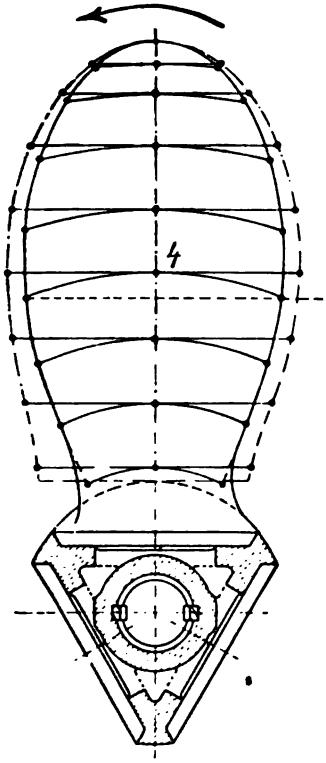


Fig. 367.

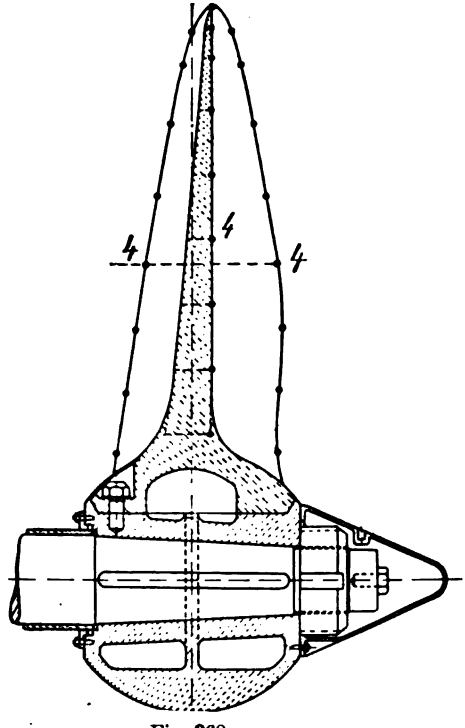


Fig. 368.

shaft may be transmitted to the boss at its thickest part, the latter must, especially at the thicker end of the cone, fit accurately on to the shaft. The boss is prevented from turning on the shaft by one or two strong keys (two are generally used in warships, although one is really all that is necessary, even for the largest screws).

Breadth of the key $b = \frac{d}{6} + \frac{5}{8}$ inch. Thickness of key $0{\cdot}5$ to $0{\cdot}6b$, d being the diameter of the propeller shaft. If there are two keys, only

$0{\cdot}8b$ instead of b is required. The keys must fit both boss and coned shaft accurately at the sides, but a little clearance may be allowed in the boss at the top. The boss is first fitted on to the shaft without the

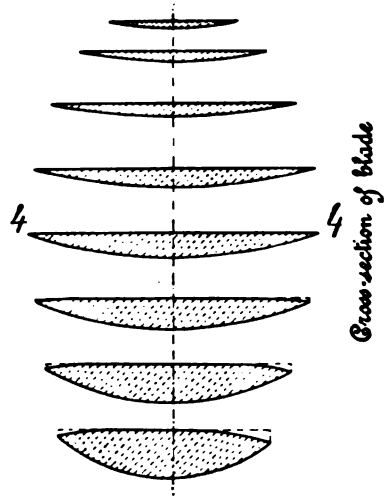


Fig. 369.

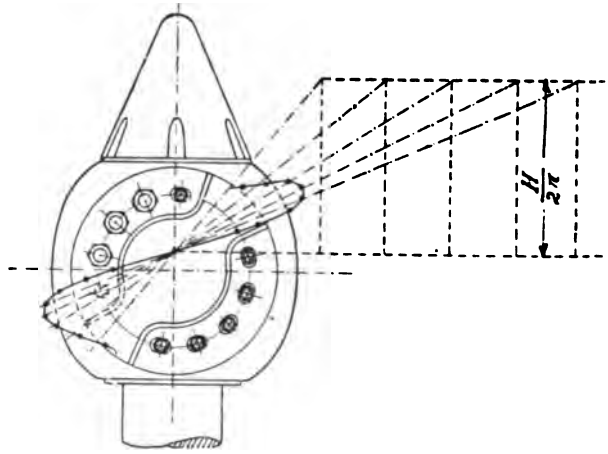


Fig. 370.

keys, then removed, and the keys fitted to the shaft in countersunk keyways. The boss is replaced, and it should be possible to push it as far up the cone as before the keys were fitted.

The keys almost always extend the whole length of the boss; but

sometimes, if the propeller is small, they occupy only the front half. The propeller nut has a fine thread, and may be made with either indentations or projections. (See Figs. 371, 372.)

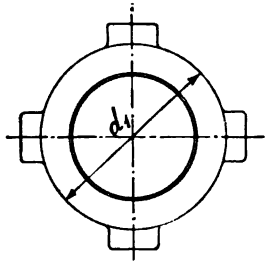


Fig. 371.

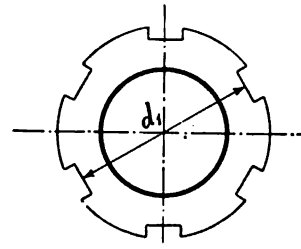


Fig. 372.

Diameter of the nut $d_1 = 1{\cdot}4$ to $1{\cdot}5d_n$. Thickness of nut $h_1 = 0{\cdot}75$ to $0{\cdot}85d_n$, d_n being the diameter over the thread. The smaller values may be used for larger nuts. These values hold for nuts where the shaft has a diameter measured outside the thread of over $5\frac{3}{4}$ inches, otherwise d_1 is taken from the table of dimensions of bolts and nuts, and equals the width across the flats of a hexagonal nut. (See Table No. 70, page 607.) To prevent the nut slacking back, it should have a left-handed thread for a right-handed screw propeller, and a right-handed thread for a left-handed screw propeller. Some method of locking the nut is also usually provided. To screw on the nut easily, the shaft is continued for a short distance beyond the nut, and given a diameter slightly less than that at the bottom of the thread.

2. *Bosses with Blades bolted on.*—In merchant vessels with propellers over 10 to 13 feet diameter, and in warships with propellers over 6 feet 6 inches to 8 feet 6 inches diameter, the blades are bolted on to the boss.

In the best practice, the flanges of the blade are very carefully fitted to the surfaces on the boss, to prevent the water getting underneath them, and sometimes a rubber ring is inserted, and screwed up against the boss.

Thickness of flange of blade $f_1 = 0{\cdot}18$ to $0{\cdot}22d$ for bronze or cast steel.

Diameter of flange of blade $D_1 = 1{\cdot}9$ to $2{\cdot}3d$.

Corresponding to this diameter of flange, the external diameter of the boss is—

$d_n = 2{\cdot}6$ to $3d$ for large screws.
$d_n = 3{\cdot}0$ to $3{\cdot}5d$ for small screws.

Length of boss with blades bolted on, $L = 2{\cdot}1$ to $2{\cdot}6d$ (higher values are for smaller bosses).

Thickness of boss round the cone—

$$c_1 = 0{\cdot}19 \text{ to } 0{\cdot}22d \text{ for bronze.}$$
$$c_1 = 0{\cdot}18 \text{ to } 0{\cdot}21d \text{ for cast steel.}$$
$$c_1 = 0{\cdot}22 \text{ to } 0{\cdot}24d \text{ for cast iron.}$$

Thickness of metal at front and back ends of boss—

$$w_1 = 0{\cdot}22d \text{ for bronze.}$$
$$w_1 = 0{\cdot}20d \text{ for cast steel.}$$
$$w_1 = 0{\cdot}24d \text{ for cast iron.}$$

In all these formulæ d is the diameter of the propeller shaft.

The blades are secured by studs fitting very tightly into the boss.

Total stud area at bottom of thread = 0·65 to 0·85 × area of the shaft.

The number and size of the studs may be more accurately calculated by taking as a basis the stress upon them due to axial thrust, turning moment, and the centrifugal force. (See "Strength of Propeller Blades.")

The number of studs should be from 6 to 10 for each blade.

Material.—The studs are best made of good steel, especially nickel steel, on account of its great strength and toughness, even when there are surface cracks. Bronze is sometimes used, but is not so reliable, even if of the strongest and best quality.

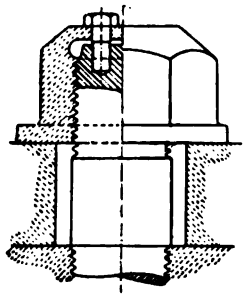


Fig. 373.

The holes in the flanges of the blade are made oval, so that the pitch can be varied; a variation of from $\frac{1}{15}$ to $\frac{1}{20}$ above and below the mean pitch should be provided for. The nuts used are generally cap nuts made of bronze, and locked in a suitable way. (See Fig. 373.)

Propeller Cap.—To protect the nut, and to reduce eddy currents, the boss of large and fast ships is often fitted with a cover of cast iron or bronze, which is screwed on to the back of the boss (Figs. 366, 368).

General Remarks.—On the back face of the boss two very large screwed holes are provided, into which bolts can be inserted, to draw off the propeller. The arrangement is shown at Fig. 374. For large propellers the hydraulic press used for dismantling the shaft couplings is also employed to draw the propeller boss. A spare boss and one or more spare blades are almost always carried, or a complete spare screw,

when the boss and blades are cast in one. The conical bore of the spare boss should be fitted as carefully as possible to the actual stern shaft, and also to the spare shaft, before it leaves the shops. In any case, for hurried repairs, it is advisable to have a template of the cone, and of the nut and key of the propeller.

To prevent sea-water penetrating between the boss and the shaft, the front end of the cone is made watertight by means of a small

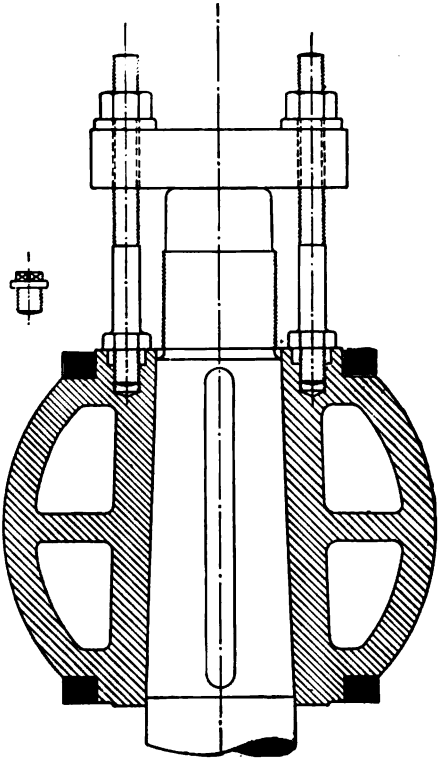


Fig. 374.

stuffing box and rubber ring, or some similar arrangement. The whole boss, where hollow, is filled in with tallow, and tallow is also forced into the clearances in the conical cap on the boss through a hole provided for that purpose. The nuts are frequently set round with cement, so that the surface of the boss may be even and continuous. In war-ships the nuts are generally covered in with a metal case, thus giving the boss the shape of a smooth ball (Fig. 368). In order to obviate any tendency to knock or shock in the screw, it is sometimes carefully

balanced so that its centre of gravity coincides exactly with the centre of the shaft. This is effected by fitting leaden weights under the blades, if they are screwed on, or at some other convenient place on the boss.

A hole is generally bored in each blade, at about one-third its length from the tip, into which an eye-bolt can be inserted, for fitting or removing the blade.

§ 234. **Machining the Surface of the Blades.***—In the latest modern practice the propeller blades of fast steamers are worked up mechanically by means of a special machine in such a way, that the surface is trued into the required mathematical shape of a helical screw. The machine consists of a horizontal arm, which swings to and fro over the blade; the centre of oscillation coincides with the propeller axis. A slide block works along this arm, on which a steel projection moves vertically up and down in a socket, so that by the combined oscillating movement of the arm, and vertical movement of the steel projection, helical lines are described.

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* German Patent 145007, *Engineering*, 1903.

PART IV.

PIPES AND CONNECTIONS.

SECTION I.

FLANGES, VALVES, ETC.

§ 235. **General Remarks.**—The pipes and connections should be kept as simple and straight as possible, and the number of flanges, hangers, valves, &c., as low as possible.

For expansion due to heat, drainage, &c., see Section III., page 423. All the piping and connections are generally tested by water pressure to double the working pressure.

§ 236. **Pipe Connections.**—For thickness of the metal of T pieces and bends see "Thickness of Valve Bodies," page 415. No pipes over $\frac{3}{4}$ inch diameter carrying high-pressure steam should have branches brazed on. Screwed joints with unions or sockets (see Fig. 375) should only be allowed with pipes under $\frac{3}{4}$ inch diameter.

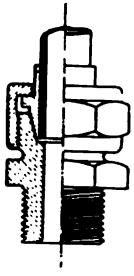


Fig. 375.

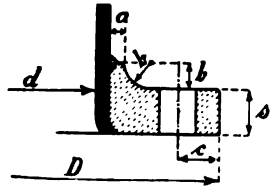


Fig. 376.

§ 237. **Flanges.**—(*Translator's Note*)—[Tables Nos. 45 to 47 in the German Edition, giving the proportions of gunmetal flanges either riveted or brazed to copper pipes, have been omitted, because if converted the equivalent English sizes would all be odd dimensions, and also because the Engineering Standards Committee are very shortly bringing out a set of Standard sizes for pipe flanges, which it is hoped will be universally adopted. These Standards will be procurable from the Secretary of the Engineering Standards Committee at 28 Victoria Street, Westminster, S.W., or from Messrs Crosby Lockwood & Son at 7 Stationers' Hall Court, London, E.C.—L. S. R.]

Gunmetal flanges are generally brazed to copper pipes, and if of large diameter may be riveted in addition. (See Figs. 377 and 380.)

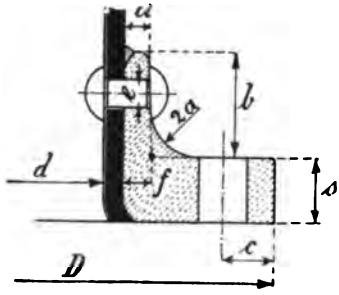


Fig. 377.

Wrought-iron flanges may be attached to wrought-iron pipes by brazing, screwing with a fine thread (see Fig. 381), or by welding, which in recent practice has been found to answer well. Joints in iron pipes may also be made by means of loose flanges. (See Fig. 382.) Gunmetal or bronze flanges must not be brazed to iron pipes, on account of their different coefficients of expansion. For lead-pipe connections see Fig. 383.

Figs. 376 and 377 are high-pressure, and 378, 379, and 380 low-pressure flanges.

§ 238. **Jointing.**—Low-pressure flanges are jointed with rubber insertion, &c.; high-pressure water pipes with rubber insertion containing fine wire gauze; steam piping with asbestos, rubber asbestos, corrugated copper rings, wire gauze and red lead, &c. Thick, flimsy packings should be avoided, as they are apt to blow out. It is important that the flanges should fit the corresponding surfaces evenly and accurately, and not be turned hollow, as otherwise leakage may take place through the bolt holes. For steam piping lenticular packing (metallic discs) is sometimes used; with this kind of packing the flanges must be strong and rigid. Pipes and connections which are exposed to great variations of temperature, *e.g.*, steam piping used alternately for saturated and superheated steam, are very difficult to keep perfectly tight. For this reason the practice of late has been to use no packing for the flanges of the H.P. steam pipes, but to make them steam tight by carefully scraping up and polishing the surfaces. Flanges where the packing is recessed (see Fig. 382) have the advantage tha it cannot blow out, but on the other hand the pipes have to be forced apart when being erected, and the flanges are thus subjected to considerable strain.

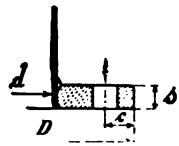


Fig. 378.

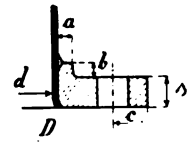


Fig. 379.

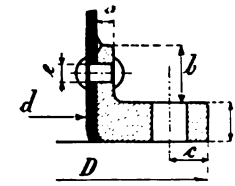


Fig. 380.

§ 239. **Bulkhead Fittings.**—Where pipes have to pass through watertight bulkheads, so-called bulkhead flanges (Fig. 384) or bulkhead stuffing boxes (Fig. 385) must be fitted.

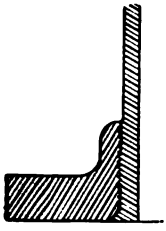


Fig. 381.

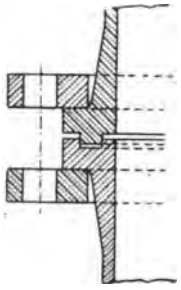


Fig. 382.

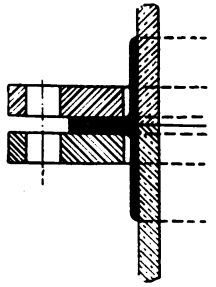


Fig. 383.

§ 240. **Extract from Regulations of German Lloyd's respecting Valves, Cocks, Pipe Connections, and Pumps.**—With the exception of the bilge and the water-ballast tank suctions, and the steam winch and safety valve exhausts, all the piping must be

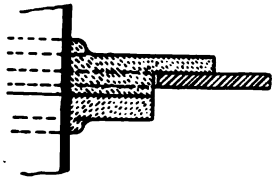


Fig. 384.

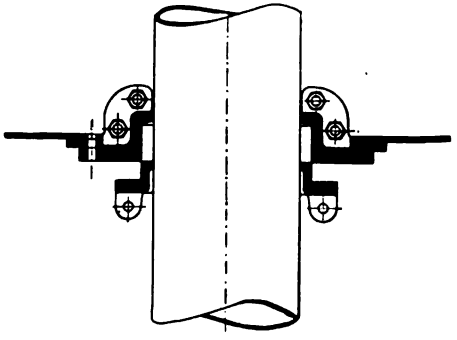


Fig. 385.

of copper, except where other material is expressly sanctioned by the Surveyor of the German Lloyd's. All valves fitted to the ship's side, and as far as possible all other valves and cocks, must be easily accessible, placed as far as possible above the flooring of the boiler and engine-room, and so arranged that no doubt can possibly exist whether they are open or closed. The heads of all bolts securing the valves and cocks to the ship's side must be countersunk.

All sea-water inlets must be provided with gratings, and also with a small steam pipe for keeping the grating clear. Wherever there is any possibility of water flooding the ship, the pipes must be fitted with two entirely independent valves, so that, even if carelessly handled, flooding

of the ship cannot take place. For the small watertight spaces in the run of the ship independent suctions are not required. The ends of the galvanised-iron suction pipes must be fitted with movable, easily accessible, galvanised mud-boxes. If the ballast-tank suction or the bilge-suction pipes are carried through the hold, they must be well protected, to prevent their being damaged when loading and unloading the cargo. Neither steam pipes nor delivery pipes should be led through the hold.

For sea-going ships, the valve seats of all pumps must be of bronze; the plungers and pump rods must have bronze rubbing surfaces, and

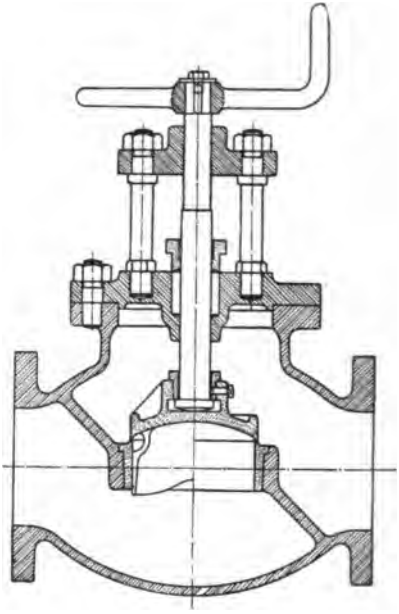


Fig. 386.

pump cylinders must have bronze liners. Feed-pump plungers or pistons may be of steel. In general all feed pumps (whether driven direct or independent) must be fitted with a safety valve, which cannot be shut off from the pump. With automatic feed pumps a safety valve is not required.

§ 241. **Valves.**—A distinction is made between "*straight-through*" or *globe valves* (see Fig. 386) and *right-angled valves*. (See Figs. 398 and 399.) As a rule the former offer more resistance to the passage of the steam or fluid than the latter, and therefore cause more loss of pressure.

Material.—For valves exposed to pressures below 180 lb. per

square inch, cast iron may be used, but it should only be employed above 130 lb. per square inch when economy is the very first consideration. In merchant ships the exhaust steam valves are always, and the bilge and ballast water valves are generally, made of cast iron. These cast-iron valves have gunmetal valves, seats, spindles, stuffing boxes, glands, and spindle nuts. Cast-steel valves are only used for steam pipes when exposed to high pressure, and they have the same gunmetal fittings as cast-iron valves. In warships cast steel is also often used for those fittings of the bilge pipes which are carried through the double bottom of the ship, or are situated immediately over it. Gunmetal valves are largely used, especially for feed-water and small steam pipes, and for the sea-water piping most liable to corrosion.

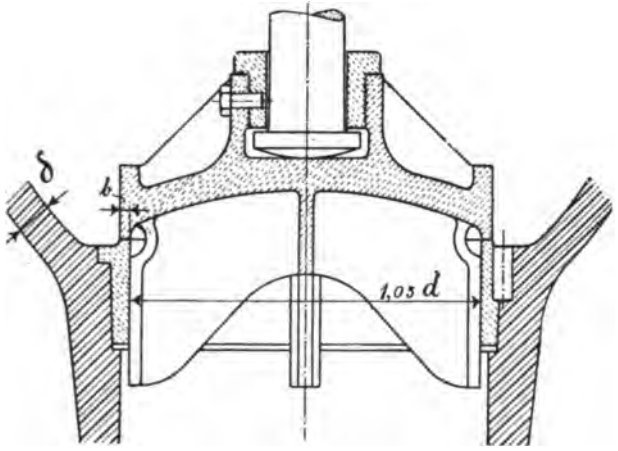


Fig. 387.

Valve Seats and Cones.—The seats are generally fitted separately, even with gunmetal valves, in order that they may be renewable. They must be conical in shape, very tightly and accurately fitted, and held in place by strong set screws. (See Fig. 387.) In spite of this, in iron or steel valves, especially where superheated steam is used, the seats, owing to unequal expansion, are apt to wear loose, and therefore of late in cast-steel steam valves, the seats are also often made of steel. They may be either conical or flat (Fig. 387). For steam they are made very narrow, for water conical and somewhat wider. Angle of cone $\alpha = 30°$ to 45° (Fig. 398). Width of seat for steam (see Fig. 387) about $b = 0{\cdot}08 + \frac{d}{100}$, d being the internal diameter of the pipe.

The *Valve* is always of bronze, generally with three or four feather guides. The spindle is almost always fitted into the valve in such a

way that the latter can move freely. (See Fig. 387.) Care must be taken that the guides are of such a length that, when the valve is open, it is guided far enough to prevent it canting, as it closes. In very large horizontal pump valves it is desirable to have a wide lip to the valve besides the ribs to guide it into the valve chest.

The *Spindles* are almost always of wrought bronze, but sometimes of steel, with a square or V thread. In bronze valves the thread is turned in the cover, and in cast-iron valves a bronze nut is fitted into the cover. (See Fig. 388.) The valve is opened by turning the spindle, or the thread on the spindle may work in a nut fitted into a support or bearing outside the valve, which is turned by a hand wheel or crank.

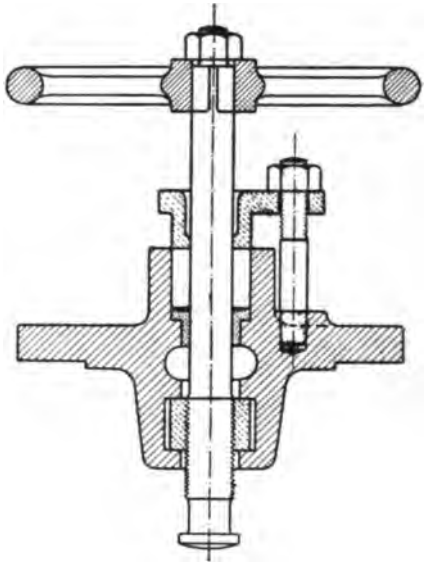


Fig. 388.

In the latter case the spindle must be prevented from turning round. With this arrangement the frictional resistance to the motion of the valve is considerably greater than where the spindle simply rotates.

The permissible stress in valve spindles of wrought bronze, either tensile or compressive, at the bottom of the thread, is as follows:—

s = 2,000 to 3,000 lb. per square inch for small, up to 6,500 lb. per square inch for large valves.

The total load must here be taken as the clear sectional area of the seat × maximum pressure in the pipe. The thread of the spindle must be so cut that the valve is closed by turning the hand wheel to the right (*i.e.*, clockwise). This must be borne in mind when arranging to

work the valves at a distance, say, from the deck, and the hand wheels, levers, &c., must be so set that they do not alter the direction of the motion. Sometimes, in very large valves, where the steam pressure falls upon the valve, the head of the spindle is made in the form of a small valve, which opens a very little as the spindle lifts, and before it carries the valve itself with it. In this way the pressure is equalised above and below the valve, and the opening of the valve is easily effected. (See Fig. 401.)

The *Lift* or amount the valve opens is generally somewhat more than a quarter the internal diameter of the pipe.

(For valve *flanges*, see Translator's Note, page 409.) In cast-iron valves less than 4 inches diameter the flanges are about 30 °/₀ and in larger valves about 25 °/₀ thicker than gunmetal flanges. Care must be taken that the bolts can be easily put in place, and especially that with right-angled valves the flanges are not too close together. The stress on the *cover bolts or studs* must be taken as the same as that on the flange bolts; and the stud holes must in no case be drilled through into the steam or water space.

The *Valve Body* must allow as free a passage as possible to the fluid (water or steam), and must therefore be of ample size round the seating.

Thickness of Valve Bodies (d = internal diameter of the pipe in inches).

Bronze high-pressure valves for feed and steam pipes $\delta = \frac{d \cdot p}{7{,}110} + \frac{3}{16}$ inch.

Bronze low-pressure valves - $\delta = \frac{d}{100} + \frac{3}{16}$ „

Cast-steel high-pressure valves
$\delta = \frac{d \cdot p}{5{,}688} + \frac{3}{16}$ „ for merchant ships.
$\delta = \frac{d \cdot p}{7{,}110} + \frac{3}{16}$ „ for warships.

Cast-iron high-pressure valves. See Table No. 51, page 433.

(*Note.*—Very large valve chests must be considerably thicker than the values given in Table No. 51.)

Cast-iron low-pressure valves
$\delta = \frac{d}{30} + \frac{1}{4}$ inch for exhaust steam pipes.
$\delta = \frac{d}{20} + \frac{1}{4}$ „ for sea-water pipes.

The above thicknesses should only be used in low-pressure cast-iron valves for pressures up to 85 lb. per square inch, and in high-pressure

valves for pressures up to 230 lb. per square inch. In light warships the walls of the valve bodies are made considerably thinner. For valve chests, see page 434. For double-seated valves, see page 141. Sea-suction and discharge valves, see Section II., page 418. Non-return and spring-loaded valves, see page 422. Safety and feed-water valves, see Boiler Fittings, page 569.

Arrangement of Valves.—All valves must be so arranged that they are easily accessible, *i.e.*, that packing can be easily inserted between the

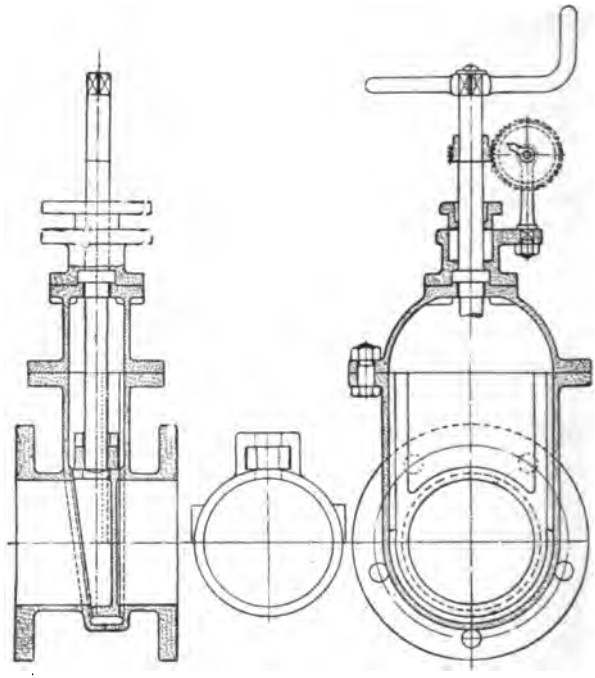


Fig. 389.

flanges, and the bolts put in place, and that the hand wheel or lever can be easily worked.

The arrangement of piping should also be as simple as possible, and on this account, for the reasons stated above, "right-angled" valves are to be preferred to "straight-through" valves. The steam should, as far as possible, be on the under side of the valve when it is shut down, in order that the stuffing box may be repacked when the valve is closed, and the valve easily opened; and lastly, that in main engine stop valves, the valve can be adjusted with ease and accuracy in spite of backlash. The valves should be grouped as far as possible, to

reduce the number of joints. Valves connected at the bottom end of a long length of vertical piping must have drain-cocks fitted above them.

§ 242. **Sluice Valves** are chiefly used for low-pressure pipes.

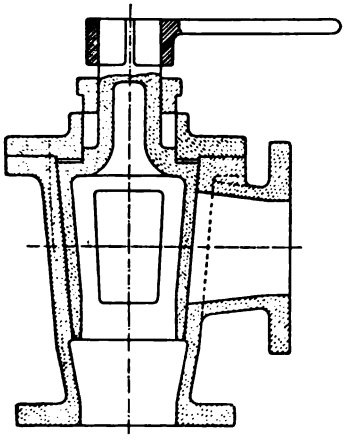


Fig. 390.

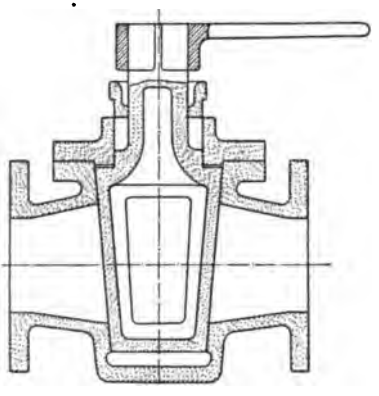


Fig. 391.

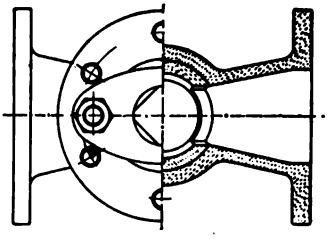


Fig. 392.

The valve bodies are in one or two parts, either of bronze or of cast iron, with gunmetal fittings; the valves themselves are nearly always of bronze. The valve may have faces on one or on both sides. Single-faced sluices have lugs or projections on the back, arranged so as to fit against corresponding lugs on the valve body, and thus press the face firmly against the seat. The spindles are of bronze, and the nuts are generally in the valve, some kind of index being provided to show how far the valve is open. The valves have guide blocks at the sides, which run in grooves in the body, so that they do not twist as they work.

§ 243. **Plug Cocks** are used almost exclusively for water pipes. For steam they are only employed as indicator cocks, blow-off cocks, auxiliary steam-cocks, and sometimes instead of reversing valves, &c. Steam-cocks of large diameter are difficult to move and seize easily, and should therefore, if possible, be avoided. Cocks are generally used for bilge pipes, lavatory pipes, and cooling water circulation, &c. They are then usually made of bronze, with stuffing boxes, and are either right-angled (Fig. 390) or "straight" or "throughway" (Figs. 391 and 392).

SECTION II.

UNDER-WATER FITTINGS.

§ 244. **Under-water Fittings.**—Under this head are included in general all those fittings which are connected to the outer skin of the ship. They lie below the water-line, and form the connection from the sea to the pipes inside the ship.

In cargo vessels the castings are mostly of cast iron, with gun-metal fittings, whereas in the navy they are always of bronze throughout. Most of the valves in merchant ships consist of ordinary valves opening inwards, but on warships the old so-called "Kingston" valves are still frequently met with. These open from within outwards, and are therefore less easily accessible for examination and repairs (Figs. 393 and 394). For a smaller Kingston valve see Fig. 395. For smaller pipe connections ordinary cocks are often used as sea-water cocks.

All openings in the outer skin of the ship, through which water enters, should be provided with gratings of bronze, wrought iron, or cast steel, the openings of which should not be more than $\frac{5}{8}$ inch wide, and their total sectional area at least 50 °/₀ more than the area of the valve itself (Fig. 393). Where single separate sea-valves are employed, as in the stokehold, they are fitted direct to the outer skin, while as a rule all those in the engine-room are connected to a cast-iron or wrought-iron chest attached to the skin of the ship, and fitted with one grating. If of cast iron, the chest is fixed to the skin by gunmetal studs, if of wrought iron it is riveted on. In the first case the hole in the skin is strengthened by means of a stiffening ring riveted to it, and the studs are only fixed into the stiffening ring. If the gunmetal sea-cocks and valves are fitted direct on to the skin of the ship, it is desirable to fit a light zinc collar at the inlet, which can be easily replaced, and which prevents the outer skin from being eaten away. (See Fig. 396.) In fitting sea-cocks care must be taken that they do not get stopped up with sand or mud. It is best therefore to have both valves and valve chests fitted above the floor line, and if possible so arranged that all the valves and cocks are visible, and can be worked without the necessity of lifting the floor plates. In any case, all under-water fittings must be so made that it can be seen at a glance whether they are open or closed.

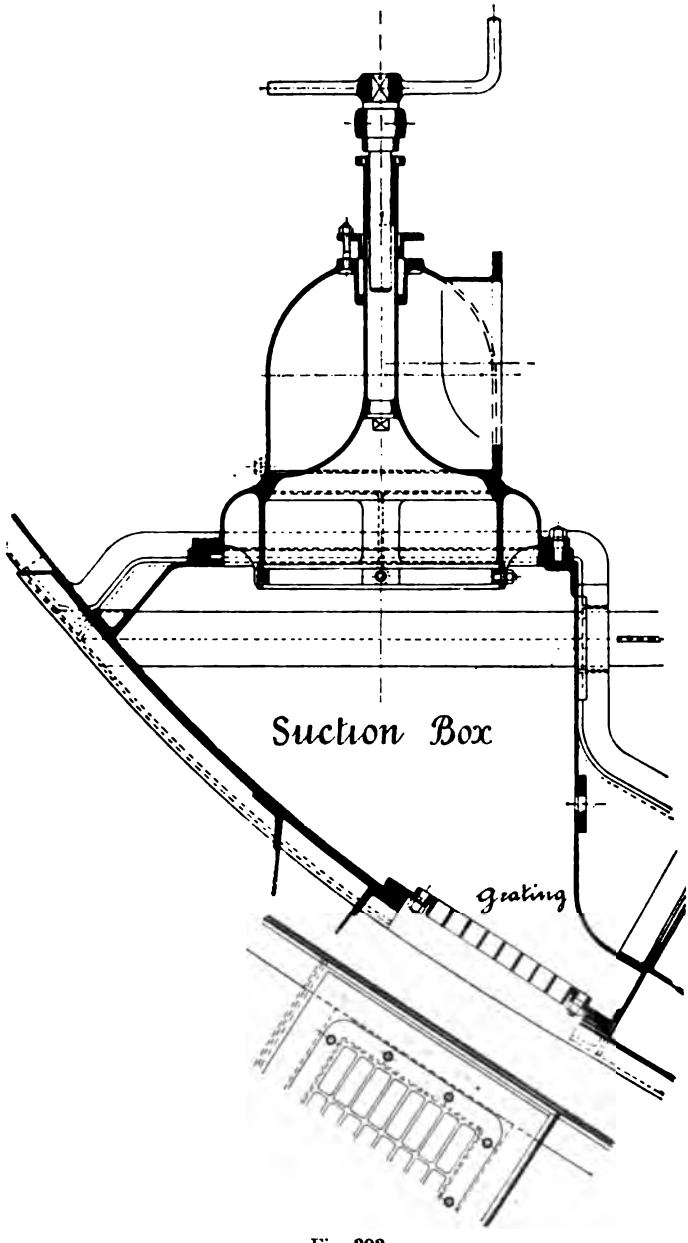


Fig. 393.

It is best to fit the gunmetal blow-off cocks for the boilers, evaporators, &c., direct to the skin of the ship, and secure them with a specially strong flange and stiffening ring. They are sometimes made with a projection which passes through the side of the ship and has a gunmetal flange fitted over it on the outside, in accordance with Lloyd's Rules (see Fig. 397). It is a better plan, however, to leave the outer skin quite smooth, and fit a zinc guard ring, which can be easily renewed, round the orifice (Fig. 396). It is not good practice to have the blow-off cocks in a cast-iron chest attached to the skin, as they are liable to

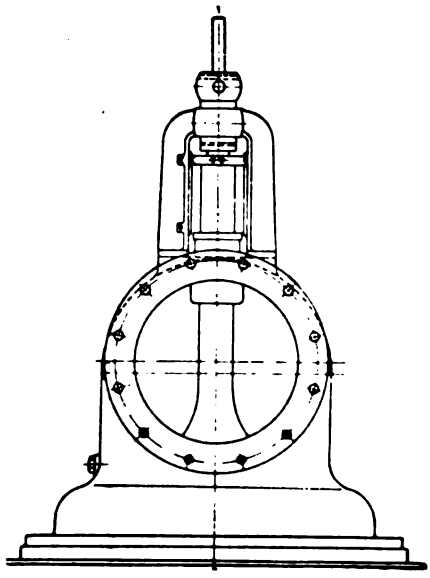


Fig. 394.

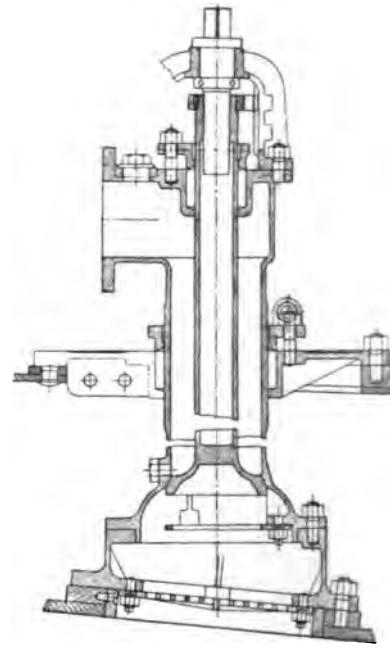


Fig. 395.

crack, owing to the sudden and unequal heating caused by the rushes of hot water. Boiler blow-off cocks must be so arranged that the keys used to work them can only be removed when the cocks are closed.

To free the gratings from ice, &c., sea-cocks are often fitted with a small steam pipe, the diameter of which should be from $\frac{5}{8}$ to $1\frac{1}{2}$ inch. It must be fitted directly to the valve chest or the neck of the valve by means of a copper pipe, through which the steam plays direct on to the gratings.

It is important that all sea-cocks be strongly constructed, especially if

of cast iron, as they are exposed to corrosion. Valves and seats should be of bronze, valve spindles of Delta metal, and the thread of the screw outside the valve box. The collar for carrying the nut of the spindle is either wholly of bronze, or of wrought iron with a gunmetal nut. It is best to make the glands and studs of the stuffing box of bronze. Sea-cocks should always have some arrangement to prevent the plug being forced out (see Figs. 390 to 392). Bolts and studs, the latter especially, for gunmetal cocks or valves should be of gunmetal, as iron bolts rust very quickly.

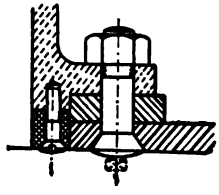


Fig. 396.

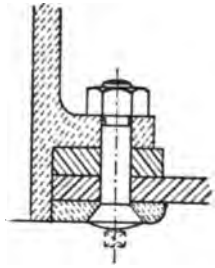


Fig. 397.

§ 245. **The Discharge Valves** or overboard delivery valves of the different pumps are generally above the water-line in merchant ships, and made of cast iron with bronze fittings, while in war vessels they are made exclusively of bronze, and generally placed below the water-line. To reduce the number of openings in the skin of the ship, some of the small discharge valves are often made

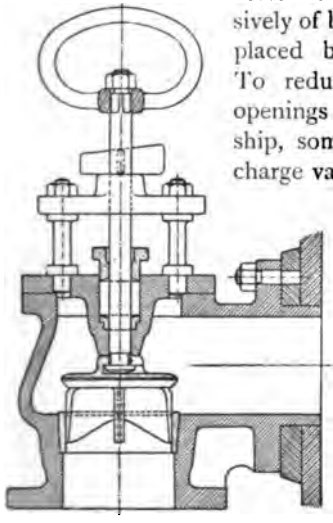


Fig. 398.

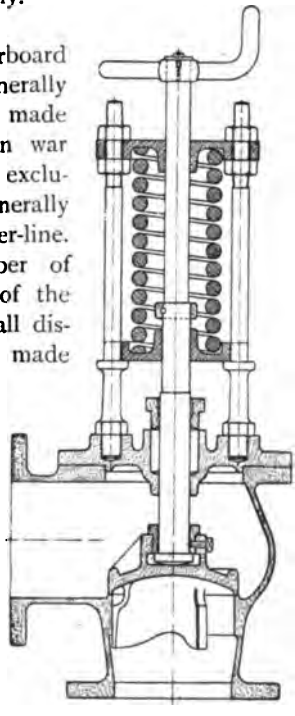


Fig. 399.

to open into the valve chest of the larger valves, the dimensions of which are correspondingly increased. All discharge valves for plunger pumps should be arranged to open outwards automatically with the

pressure of the water, and be held in position when open, but not when closed. (See Fig. 398.) This is of special importance with the discharge valves of the air, circulating, and bilge pumps attached to the main engine. Discharge valves serving these pumps are frequently loaded with a spring (see Fig. 399), but this arrangement is not really necessary for the discharge valves of centrifugal pumps.

To avoid difficulties in working the air-pump discharge valves of engines, which, although generally worked surface condensing, can also be worked jet condensing, these valves may be held closed by a carefully adjusted spring, allowing the valve to open automatically in case of need.

Discharge valves, like sea-cocks, are secured to a stiffening ring riveted to the skin of the ship, by studs, or bolts with coned heads. As a rule these valves have a spigot passing through the outer skin, and ending flush with it on the outside. If they are above the water-line, the studs securing them may be of wrought iron; if they are below the water-line, or are made of bronze, gunmetal studs must be used. All discharge valves must be so arranged as to be accessible at all times. If they cannot be fitted direct to the outer skin, on account of the coal bunkers at the side or other impediments, a strong wrought-iron pipe may be fixed between the skin and the valve. Where it passes through the coal bunkers, &c., this pipe must be protected from injury by a stout wood or metal covering.

SECTION III.

MAIN STEAM, AUXILIARY STEAM, AND EXHAUST PIPING.

1. Main Steam Piping.

§ 246. **Main Steam Piping.**—If there is only one boiler, the main steam pipe leads from the main stop valve on the boiler direct to that on the H.P. valve chest of the engine. If there are several boilers, the pipes from each boiler are generally connected, and only *one* main steam pipe takes the steam to the engine. In the case of engines of large power (especially on warships) working at high pressures, instead of one large pipe there are sometimes several smaller ones leading from the boilers to the engines, which are only connected up when they reach the engine-room bulkhead. Smaller diameters, reduced thickness of the piping, and greater safety in working are thus obtained. If there are two engines, it is better that each should have its own separate main steam pipe. Generally each of the above pipes can be shut off, if necessary, at the forward engine-room bulkhead, and the port and starboard pipes are connected at that point by a junction pipe which can be shut off at will. If this cannot be done, any break which may occur must be isolated by means of a blank flange.

§ 247. **Draining of Steam Pipes.**—To avoid what is known as "water-hammer," the steam pipe leading from the boiler stop valve to the engine is frequently given either a slope downwards or upwards. With the latter arrangement any condensed water can, if the engine is stopped, flow back to the boiler; with the second it collects in front of the stop valve at the engine, or in the *separator*. Bends in pipes where any water can collect, or what are known as "water-pockets," must be strenuously avoided, or if this is impossible, they should be fitted with drain-cocks. In short steam pipes a *separator* (Figs. 400, 401) is not absolutely necessary, but there should be one in every longer steam pipe as near the engine as possible, to separate the condensed water and also the priming water from the steam. If there is no separator, it should be replaced by a drain-cock at the lowest part of the pipe, and as close as possible to the engine. The condensed water can be led either into the condenser or into the hot well. The latter is the better plan, because it helps to heat the feed-water. To ensure continuous drainage, *automatic traps* are sometimes fitted, as well as a separator,

but they do not as a rule work satisfactorily. It is generally sufficient to draw off the condensed water by hand at intervals, or so to adjust the separator drain-cock that the level of water in the separator is always uniform. Sometimes a special vessel fitted with a gauge glass is provided below the separator, from which the water is discharged.

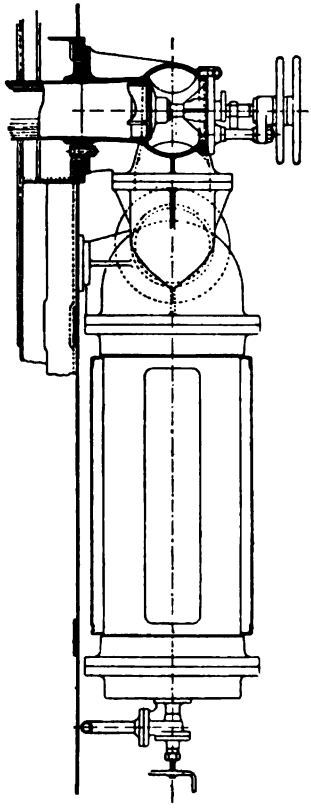


Fig. 400.

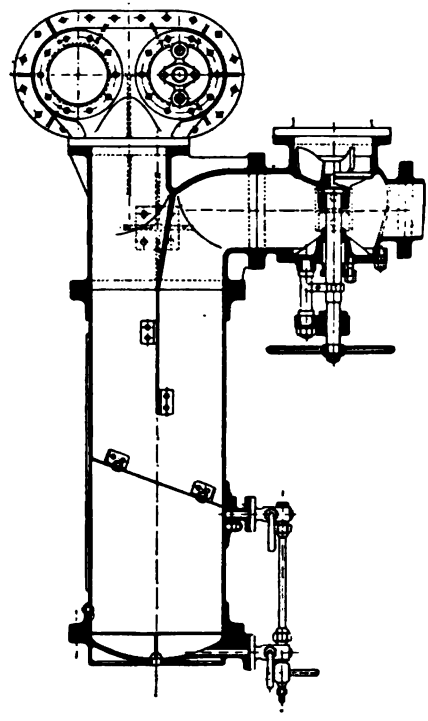


Fig. 401.

§ 248. **Diameter of the Steam Piping.**—The internal diameter is calculated to give a mean speed of the steam through the piping of from 6,000 to 8,000 feet per minute. (Compare page 128.) The internal area of main steam pipes is generally from 8 to 10 °/₀ less than the sum of the areas of the branch or contributory pipes from each of the boilers, as the frictional resistances are proportionately less in the larger pipes. In the case of long steam pipes, slightly larger diameters should be allowed than for shorter pipes. Loss of pressure from the boilers to the valve chest of the H.P. cylinder should not exceed from 4 to 8 lb.

§ 249. **Expansion due to Heat.**—*The steam pipes must be so arranged that they can expand and contract freely.* To effect this the piping must have large *expansion bends* or *expansion joints.* In steam pipes up to 5 inches internal diameter in which sufficient bends already exist, special arrangements to allow of expansion are superfluous. Should, however, the pipes be *straight*, expansion joints must be fitted. Care must also be taken that the expansion of the pipes is not interfered with by the hangers and supports securing the pipes to the bulkheads and decks, as any obstruction to free expansion may set up dangerous strains in the pipes and flanges; for this reason the pipes are only fixed in a few places, and allowed to slide elsewhere loosely in the hangers.

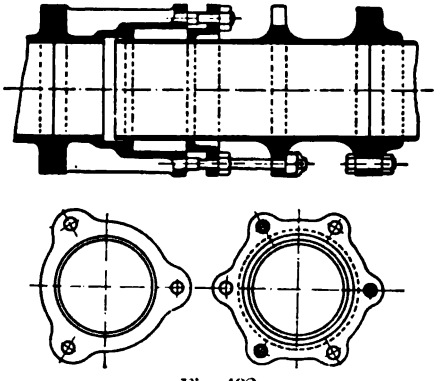


Fig. 402.

If the pipes are coiled to allow for expansion due to heat, it is advisable to calculate the vertical height of the coil according to the following formula—

$$A = 2\sqrt{d \times L}$$

Where A = vertical height of the coil in inches.
L = distance between the two fixed ends of the coil in inches.
d = external diameter of the pipe in inches.

For iron pipes it is best to have the short bends of steel. These should not, however, be used to take up the expansion of the pipes due to heat, but provision should be made for this by means of a stuffing box or expansion joint.

If expansion joints (Fig. 402) are used, care must be taken that the pipes can only move in the desired direction.

Those parts of the piping which are assumed to be fixed must be sufficiently strong to withstand the strains thrown upon them, and special care must be taken to prevent any movement in them. Bulkheads, deck beams, &c., utilised to support the pipes, must be strengthened (Fig.

403). Stop valves, junction pieces, &c., on the boilers when fitted with expansion joints must be strengthened on the boiler side, in order that *bending strains on the neck of the valves may be avoided.* The gland of the stuffing box is also often secured with stays to the other end of the pipe, so that the thrust may be taken up by these stays (Figs. 404-406). In pipes where bends are employed, the strains upon the supports can only be approximately determined.

In expansion joints the power acting upon the ends of the pipes tending to make them move in an axial direction, that is, to force the ends of the pipes out of the joint, is $P = \frac{D^2\pi}{4} \cdot p$. P being the thrust in pounds, D the *external* diameter in inches of the pipe in the

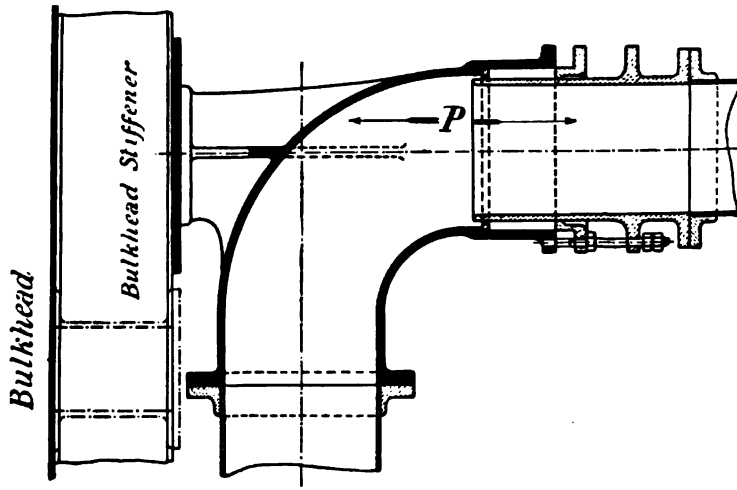


Fig. 403.

stuffing box, and p the steam pressure in the pipe in pounds per square inch above atmosphere. The friction of the stuffing box is here neglected; this friction may, if the vibrations of the ship are considerable, produce an extra tensile or compressive stress. The use of the above formula gives quite sufficiently accurate results. The end of the pipe which slides in the stuffing box is generally of bronze, and is joined to the pipe itself by a flange (Figs. 402, 403).

To prevent it from being blown out of the stuffing box when the pipe yields to the thrust P, ties or safety bolts are often fitted, which connect the stuffing box to a fixed flange on the pipe. Sufficient play must be allowed, when hot, between the movable flange and nuts of these tie bolts, that when the pipe cools it does not throw a tensile strain upon the bolts. The latter are fitted with lock nuts. These

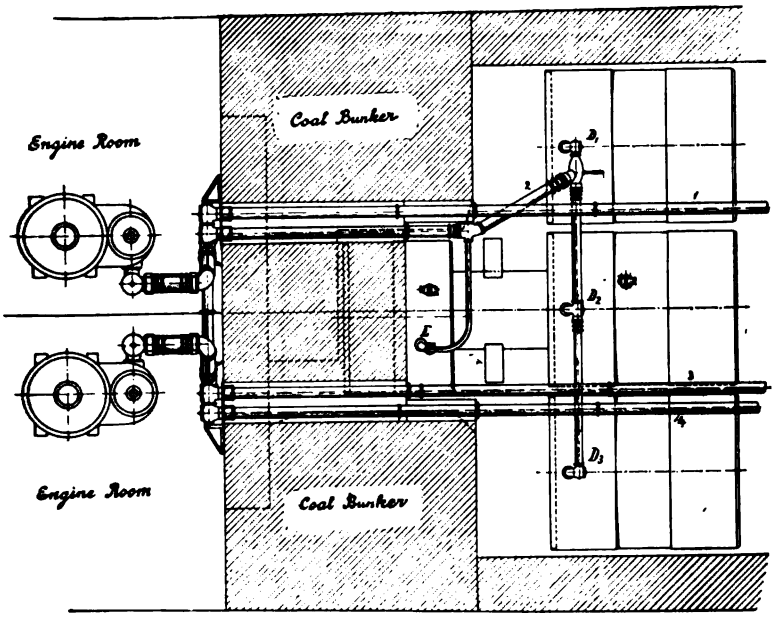


Fig. 404.

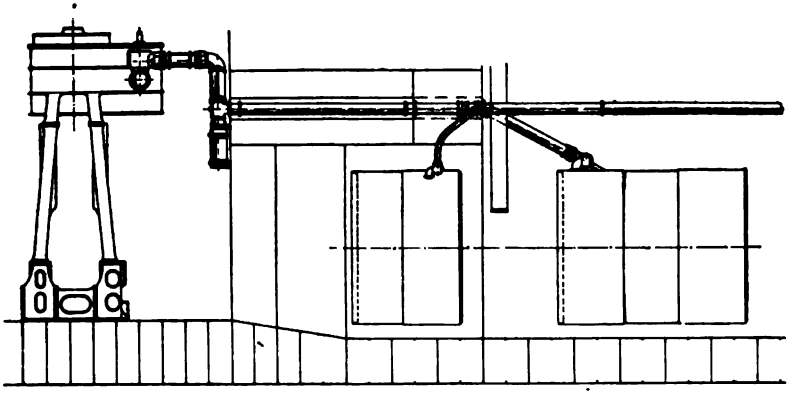


Fig. 405.

safety bolts frequently form at the same time the studs for the stuffing box (Fig. 403); it is best to use bronze nuts, as they do not rust, and are easily screwed up even when hot.

The *expansion of pipes* per foot of length from 32° F. up to the temperature of the steam is as follows:—

Material.	Steam Pressure—Pounds per sq. in. above Atmosphere.				
	100 lb. per sq. in.	130 lb. per sq. in.	160 lb. per sq. in.	200 lb. per sq. in.	230 lb. per sq. in.
	Inch.	Inch.	Inch.	Inch.	Inch.
Copper - - -	0·03	0·033	0·035	0·037	0·038
Wrought iron, steel, or cast iron -	0·02	0·023	0·024	0·025	0·026

§ 250. **For Arrangement of the Main Steam Pipes** in a large passenger steamer see Figs. 404 to 406. The pipes marked 1, 3, and 4 convey the steam from the forward group and the two central groups of boilers, while pipes marked 2 take the steam from the aftermost group of boilers, consisting of three double-ended and one single-ended cylindrical boiler. The pipes 1 and 2 on the port, and 3 and 4 on the starboard, are all connected to a cast steel valve chest at the forward engine-room bulkhead. From here the steam passes to the steam separator below, and thence through a main stop valve to the main steam pipe to the engine. The two port and starboard steam separators are connected by a pipe. To allow for free expansion there are three stuffing boxes in the pipe, between the steam separators and the engine. To take up the horizontal thrust along the pipe, the bend in it is stayed, not only to the stop valve on the steam separator, but also to the main stop valve on the H.P. valve chest. The stop valves are fixed to the engine-room bulkheads, which are specially stiffened; the boiler stop valves are stayed against the boiler shells; and the tee piece into which the bend from the single-ended boiler leads is similarly secured to the bulkhead and deck.

§ 251. **Thickness of Steam Pipes.**

1. THICKNESS OF COPPER PIPES.

1. Brazed copper pipes $\delta = \frac{d \cdot p}{5,688} + \cdot 06$ inch.*

2. Seamless drawn copper pipes $\delta = \frac{d \cdot p}{5,688} + \cdot 04$ inch.†

* If very solidly made $\delta = \frac{d \cdot p}{5,688} + \cdot 08$ in. † If very solidly made $\delta = \frac{d \cdot p}{5,688} + \cdot 08$ in.

For drawn copper pipes (bound with steel wire) from 4 to 16 inches the following formula is frequently used :—

$$\delta = \left(\frac{p-2}{5{,}688}\right)d + 0{\cdot}08$$

For copper pipes (bound with steel wire) above 16 inches internal diameter—

$$\delta = \frac{p \times d}{5{,}688}$$

d being the inside diameter of the pipe in inches, p the steam pressure in pounds per square inch above atmosphere, and δ the thickness of the pipe in inches.

The thickness of copper pipes which have to be bent should be from 0·02 to 0·04 inch more than the above, but the smallest mean radius of

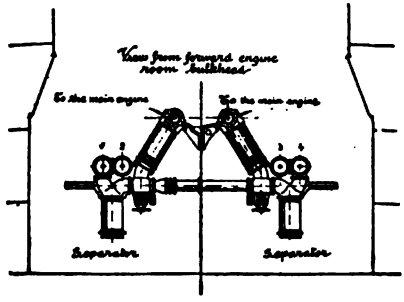


Fig. 406.

the curve must not be less than $4d$. If the diameter of these pipes is more than 4 inches, they are frequently strengthened by winding thin steel wire (about No. 11 S.W.G.) round them. This must be so arranged that if the wire is broken anywhere only a few coils can get loose. On account of their greater cost and lower tensile strength copper pipes are now being more and more superseded by iron and steel pipes. Brazed copper pipes are now seldom used for main steam pipes. Wrought iron pipes are welded, steel pipes are seamless drawn or rolled ("Mannesman" tubes).

3. Thickness of welded iron pipes $\delta = \frac{d \cdot p}{7{,}110} + {\cdot}12$ inch.

4. Thickness of seamless steel pipes $\delta = \frac{d \cdot p}{9{,}954} + {\cdot}10$ inch.

TABLE No. 48.

Thicknesses of Solid Drawn and Brazed Copper Steam Pipes.

(Calculated from Formulæ 1 and 2.)

Internal Diameter.	130 lb. Pressure above Atmosphere.		160 lb. Pressure above Atmosphere.		185 lb. Pressure above Atmosphere.		230 lb. Pressure above Atmosphere.	
	Solid Drawn.	Brazed.	Solid Drawn.	Brazed.	Solid Drawn.	Brazed.	Solid Drawn.	Brazed.
Inches.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.
1	0·068	0·088	0·068	0·088	0·078	0·098	0·078	0·098
1¼	0·068	0·088	0·078	0·098	0·078	0·098	0·088	0·108
1½	0·068	0·088	0·078	0·098	0·088	0·108	0·098	0·118
1¾	0·078	0·098	0·088	0·108	0·098	0·118	0·108	0·127
2	0·078	0·098	0·098	0·118	0·108	0·127	0·118	0·137
2¼	0·088	0·108	0·098	0·118	0·108	0·127	0·127	0·147
2½	0·088	0·108	0·108	0·127	0·118	0·137	0·137	0·157
2¾	0·098	0·118	0·108	0·127	0·127	0·147	0·147	0·167
3	0·098	0·118	0·118	0·137	0·137	0·157	0·157	0·177
3¼	0·108	0·127	0·127	0·147	0·137	0·157	0·169	0·187
3½	0·108	0·127	0·127	0·147	0·147	0·169	0·177	0·196
4	0·118	0·137	0·137	0·157	0·157	0·177	0·187	0·206
4½	0·137	0·157	0·157	0·177	0·187	0·206	0·216	0·236
5	0·137	0·157	0·169	0·187	0·196	0·216	0·236	0·256
5½	0·147	0·169	0·177	0·196	0·206	0·226	0·246	0·265
6	0·169	0·187	0·196	0·216	0·226	0·246	0·275	0·295
6½	0·177	0·196	0·206	0·226	0·246	0·265	0·295	0·314
7	0·187	0·206	0·216	0·236	0·256	0·275	0·305	0·324
7½	0·187	0·206	0·226	0·246	0·265	0·285	0·314	0·334
8	0·206	0·226	0·246	0·265	0·285	0·305		
8½	0·216	0·236	0·256	0·275	0·305	0·324		
9	0·226	0·246	0·265	0·285	0·314	0·334		
9½	0·236	0·256	0·275	0·295	0·324	0·344		
10	0·236	0·256	0·285	0·305	0·334	0·354		

TABLE No. 49.

Thicknesses of Lap Welded Wrought-iron Steam Pipes.

(Calculated from Formula 3.)

Internal Diameter.	Pressures above Atmosphere.			
	130 lb. per square inch.	160 lb. per square inch.	185 lb. per square inch.	230 lb. per square inch.
Inches.	Inch.	Inch.	Inch.	Inch.
2	0·147	0·157	0·169	0·187
2¼	0·157	0·169	0·177	0·187
2½	0·157	0·169	0·177	0·196
2¾	0·157	0·177	0·187	0·196
3	0·169	0·177	0·196	0·216
3¼	0·177	0·187	0·196	0·216
3½	0·177	0·187	0·206	0·226
3¾	0·177	0·196	0·206	0·226
4	0·187	0·196	0·216	0·236
4½	0·196	0·216	0·226	0·256
5	0·196	0·216	0·246	0·275
5½	0·206	0·226	0·246	0·285
6	0·216	0·246	0·265	0·305
6½	0·226	0·256	0·275	0·314
7	0·236	0·256	0·285	0·334
7½	0·236	0·265	0·295	0·344
8	0·246	0·275	0·305	0·354
8½	0·256	0·295	0·324	0·374
9	0·265	0·305	0·344	0·403
10	0·285	0·324	0·364	0·423
11	0·295	0·334	0·383	0·452
12	0·324	0·374	0·423	0·492
13	0·334	0·383	0·442	0·521
14	0·344	0·403	0·462	0·541
15	0·354	0·413	0·472	0·570

TABLE No. 50.

Thicknesses of Solid Drawn Steel Steam Pipes.

(Calculated from Formula 4.)

Internal Diameter.	Pressures above Atmosphere.			
	130 lb. per square inch.	160 lb. per square inch.	185 lb. per square inch.	230 lb. per square inch.
Inches.	Inch.	Inch.	Inch.	Inch.
2	0·127	0·127	0·137	0·147
2¼	0·127	0·137	0·137	0·147
2½	0·127	0·137	0·137	0·157
2¾	0·127	0·137	0·147	0·157
3	0·137	0·147	0·157	0·169
3½	0·137	0·147	0·157	0·169
3½	0·137	0·147	0·157	0·177
3¾	0·137	0·157	0·169	0·177
4	0·147	0·157	0·169	0·187
4½	0·157	0·169	0·177	0·196
5	0·157	0·169	0·187	0·206
5½	0·157	0·177	0·196	0·216
6	0·169	0·187	0·206	0·236
6½	0·177	0·196	0·216	0·246
7	0·177	0·196	0·216	0·246
7½	0·187	0·206	0·226	0·256
8	0·187	0·206	0·236	0·265
8½	0·196	0·216	0·246	0·285
9	0·206	0·236	0·256	0·305
10	0·216	0·246	0·275	0·314
11	0·226	0·256	0·285	0·334
12	0·246	0·275	0·314	0·364
13	0·256	0·295	0·324	0·383
14	0·265	0·305	0·334	0·403
15	0·275	0·314	0·354	0·413

The minimum working stress on the material (which for welded pipes, according to formula 3, is about 3,500 lb. per square inch) must be taken, because under certain conditions the pipes are apt to corrode, and moreover they cannot be examined internally. In large pipes the welded seam is sometimes strengthened with a butt strap riveted on. As considerable difficulty is experienced in bending thick pipes, wrought-iron and steel pipes of over say 3 inches in diameter are, where possible, only used in straight lengths, separate bends of cast steel being provided where bends are necessary.

The thicknesses of cast steel or bronze *pipe bends* and *tee-pieces* may be determined from the following formula:—

(1.) Cast steel bends (on account of difficulties in casting)—

$$\delta = \frac{d \cdot p}{7{,}110} + \cdot 2 \text{ inch for warships; and}$$

$$\delta = \frac{d \cdot p}{5{,}680} + \cdot 2 \text{ inch for merchant vessels.}$$

(2.) Bronze bends— $\delta = \frac{d \cdot p}{7{,}110} + \cdot 2$ inch.

In warships the thickness of the bronze may be even less.

Cast-iron pipe bends are only used for pressures up to about 185 lb. per square inch above atmosphere. For thicknesses of cast-iron valves and bends for steam piping see following table:—

TABLE No. 51.

Thicknesses of Cast-iron Valves and Bends for Steam Pipes.

Internal Diameter.	100 lb. per square inch.	130 lb. per square inch.	160 lb. per square inch.	190 lb. per square inch.
Inches.	Inch.	Inch.	Inch.	Inch.
2	0·35	0·39	0·43	0·47
4	0·47	0·51	0·55	0·59
6	0·55	0·59	0·67	0·70
8	0·63	0·66	0·74	0·78
10	0·66	0·74	0·82	0·90
12	0·70	0·78	0·90	0·98
14	0·74	0·86	0·98	1·06

§ 252. **Lagging.**—To avoid as far as possible *losses of heat due to radiation* from the steam piping, these pipes are coated with a non-conducting material, such as fossil meal, asbestos, cork, felt, &c., with an outer covering of sail-cloth, sheet iron, or other suitable material. The bends and tee-pieces are sometimes covered with lead sheeting. The flanges are generally left bare, or only provided with a light and easily removable covering of sheet metal or asbestos.

2. Auxiliary Steam Piping.

§ 253. **Auxiliary Steam Piping.**—To drive the various pumps, &c., in the engine and boiler rooms auxiliary steam pipes are used, which take steam direct from the boiler through a separate valve. If there are several boilers in the ship, each boiler has generally a sufficiently large stop valve to supply all the auxiliary steam, but where there are a very large number of boilers the auxiliary steam pipes are generally connected only to a few of them. It is best to fit the stop valves supplying these pipes with loose valves, not attached to the spindles, so that, should the wrong valve be opened by mistake, steam from the auxiliary steam pipes cannot find its way into any boiler which is laid off or not working. In general the same rules hold good for the auxiliary as for the main steam pipes.

Branch pipes leading from the main auxiliary steam pipe to the different pumps, &c., must be connected to them by cast-iron, bronze, or cast-steel tee pieces or bends. With the high steam pressures now in vogue brazed copper bends are not strong enough. Each branch pipe is generally fitted with a stop valve where it branches off at the main. Cocks should not be used in these pipes, as they are never perfectly tight, and are often difficult to work. At suitable places in the auxiliary steam piping it is well to have distributing valve chests, from which the pipes are led to the various auxiliary engines; by this means the number of flanges required is considerably reduced. Besides the connection of the auxiliary steam pipe to the boiler, it is sometimes also connected in the engine-room to the main steam piping. This arrangement is, however, only suitable in ships in which the auxiliary engines for the centrifugal pumps, steam feed-water pumps, evaporators, &c., have to be kept at work while the main engines are running.

When arranging the piping to the steam whistle, special care must be taken that the water condensed in the pipes can flow back into the boiler, and cannot collect anywhere in pockets. Drain-cocks or valves should be provided at all points in the auxiliary steam pipes where water can accumulate when the auxiliary engines are not working. This water may either be discharged into the bilge, into a condensed water tank fitted in the bilges, or into the auxiliary engine exhaust pipe. The internal diameter of the auxiliary steam pipe depends upon the size and number of auxiliary engines at work at the same time. It is only necessary to make the area of this pipe from 0·66 to 0·75 of the total area of all the branch pipes supplying steam to the auxiliary engines.

3. Exhaust Steam Piping.

§ 254. **The Main Exhaust** of large engines is led direct into the main condenser. In small engines on river steamers the exhaust is frequently led direct to the atmosphere, or may be arranged to discharge either to the atmosphere or to a jet condenser at will.

§ 255. **The Auxiliary Engine Exhausts** are generally led into one common pipe. This exhaust may be led either into the

Main condenser,
Auxiliary condenser (if there is one),
Feed-water heater (if there is one),
Or into the open air.

It should be possible to shut off each of the branch pipes by a valve or cock. One of these valves, generally the one exhausting into the open air, should take the form of a safety valve, to prevent the pressure of steam in the exhaust becoming too high through careless management. The valve should be loaded to a pressure of about 30 lb. per square inch. The exhaust from the steering engines and from the winch for raising the anchors is often led, not only into the general exhaust pipe, but also, alternatively, direct into the open air. The exhaust of the electric light engines should be taken directly into the main or auxiliary condenser, so that the steady running of these engines may not be affected by the varying steam pressure in the exhaust pipe. For use in case of emergency a connection to the auxiliary engine exhaust pipe may also be provided. The exhaust pipes of the different auxiliary engines should be so arranged that each engine can be shut off from the auxiliary exhaust. The drainage from the auxiliary engines is sometimes led into the auxiliary exhaust. For the blow-off pipe to the safety valve see page 552.

§ 255A. **The Diameter of the Exhaust Pipes** of the various engines should be about 1·2 × diameter of the steam pipe; the diameter of the main auxiliary exhaust pipe may be taken at slightly less than that corresponding to the total area of the pipes leading into it.

§ 255B. **Thickness of Exhaust Pipes.**

Up to 2 inches	(internal diameter)	= 0·06	inch.
From 2 to 4 inches	(„ „)	= 0·08	„
„ 4 to 8 „	(„ „)	= 0·10	„
Beyond 8 „	(„ „)	= 0·12	„

The main exhaust piping should, if possible, be arranged with a fall towards the condenser, and any unavoidable water-pockets must be provided with drain-cocks.

SECTION IV.

FEED-WATER PIPES.

§ 256. **Boiler Feed Pipes.**—These serve to supply *feed-water to the boilers.* As each boiler installation must, according to Lloyd's Rules, be provided with two entirely independent feed-water systems, it is usual to have also two quite independent feed-water pipes leading to separate feed-check valves on each boiler. The pipe which, under ordinary working conditions, conveys the feed-water to the boilers is generally called the *main feed pipe*, and that through which the supplementary supply can be fed into the boilers from the donkey feed pump or an injector is called *the auxiliary feed pipe.* The two pipes are often so interconnected by valves that if the supplementary supply is damaged the main feed pumps can be utilised, and the engine or main feed-water pumps can deliver through the auxiliary feed pipe. Formerly the feed-water pipes were usually placed below the floor plates of the engine and boiler rooms, but in modern practice it is customary to fix them at about the level of the upper part of the boiler, so that the flanges and pipes may be the more easily kept in view.

§ 257. **General Arrangement.**—1. In the oldest and simplest arrangement of the main feed-water pipes, the feed pumps, driven from the engine, drew the hot water from the hot well, and delivered it through the main feed pipe direct to the boiler. In later practice this method is only usual in small engines up to about 300 I.H.P.

2. In larger engines a feed-water heater (see page 445) and sometimes also a feed-water filter (see page 439) are fitted into the main feed pipe between the engine feed pump and the boiler.

3. In merchant ships with engines indicating more than 1,000 H.P. the arrangement shown in Fig. 407 is frequently met with. The feed pumps connected to and driven by the main engines deliver the feed-water drawn from the hot well into a Weir (see Fig. 415), Blake, or other type of feed heater fitted as high up as possible in the engine-room, where the feed-water is heated by the auxiliary exhaust steam. From the heater the feed-water passes to an automatic duplex steam pump

on the floor of the engine-room, which discharges the heated water through the main feed pipe to the boiler. The speed of the auto-

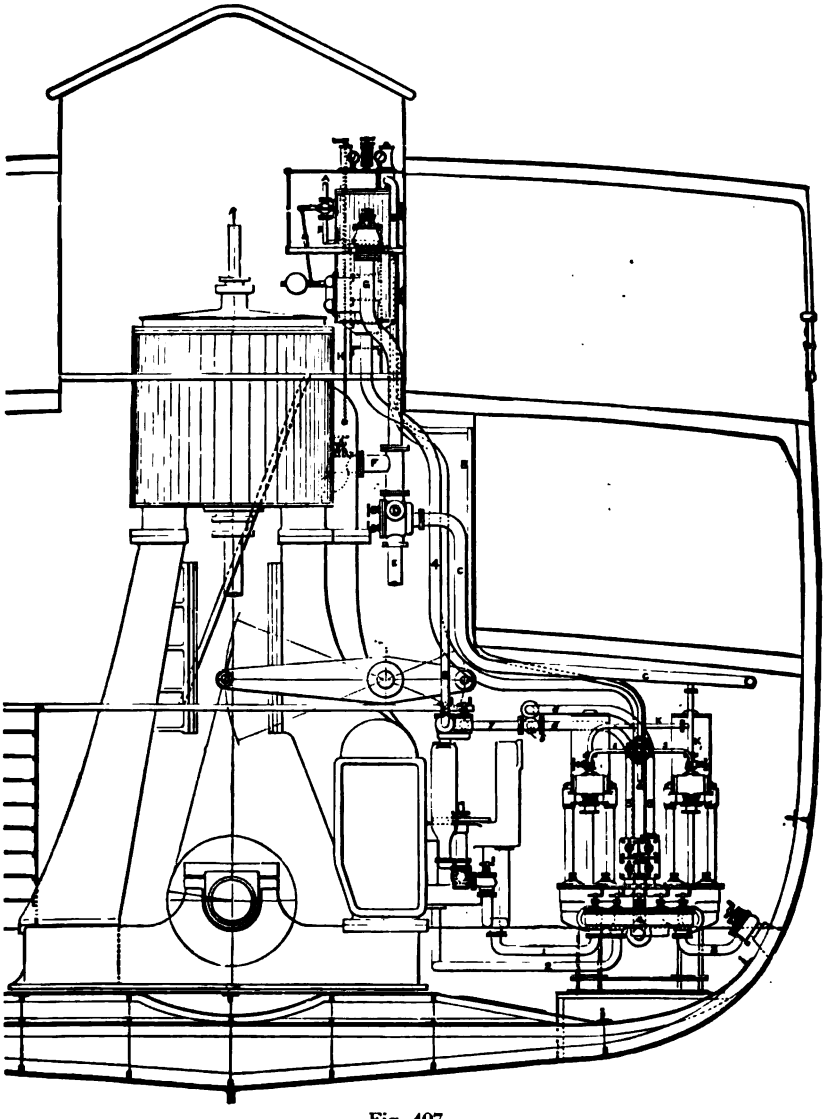


Fig. 407.

matic feed pump is regulated by a float in the lower part of the feed heater, which controls the cock supplying steam to the feed pump.

In Fig. 407—

1 = steam feed pump suction from hot well.
2 = " " " condenser.
3 = " " " sea.
4 = " " " feed heater.
5 = auxiliary feed pipe.
6 = main feed pipe.
7 = feed pump delivery pipe to auxiliary feed pipe.
8 = " " feed heater.
A = steam from the boiler or auxiliary steam pipe to the regulating cock on feed heater.
a = steam direct from boiler or auxiliary steam pipe to feed pumps.
B = steam from the regulating cock on the feed heater to the feed pumps.
C = auxiliary engine exhaust.
E = " " to condenser.
F = exhaust from auxiliary engines to the L.P. valve casing.
G = auxiliary exhaust to feed heater.
H = air pipe to condenser.
J = steam to feed pumps.
K = feed pumps exhaust.

In the arrangement shown the two feed pumps may be worked at the same time, or only one may be worked, the other being kept in reserve, and not used under ordinary circumstances. The feed-water filter can be fitted either in the feed-pump delivery pipe 8 from the engine to the heater, or in the main feed pipe 6.

4. In the feed pipe arrangements of many of the most recent vessels, only the air pump is driven from the main engines, and there are no main engine-driven feed pumps. The air pump delivers the condensed water into a so-called "float tank" (see page 441), from which the two steam-driven feed pumps draw. Under ordinary working conditions, however, only one of these pumps draws from this tank, its speed being regulated by the float. The water is forced into a jet feed heater, which, as mentioned under heading 3, is fixed as high in the engine-room as possible. The feed-water is heated by means of the auxiliary engine exhaust. The second pump then draws the heated feed-water from the heater, and delivers it into the main feed pipe, the speed of the pump being regulated by the float in the feed heater. With this arrangement a filter can be fitted either in the air-pump discharge to the hot well, in

the feed-pump delivery to the feed heater, or in the main feed pipe. Instead of driving the air pump from the main engine it may be separated from it, and worked as an independent pump, on the Blake, Weir, Worthington, or other system.

5. With air pumps separated from the main engine the water may be sent direct into a Weir's feed heater, fitted high up in the vessel, from whence the steam-driven feed pump draws it (the speed of the pump being regulated by a float), and delivers it through a filter into the boiler.

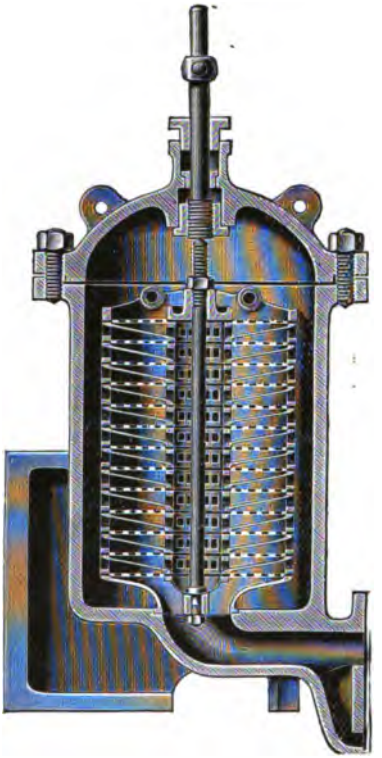


Fig. 408.

6. In warships each air pump generally discharges into a hot well, from which the feed pumps in the boiler-room draw their supply.

This hot well is sometimes also arranged to act as a feed-water heater and purifier, the steam from the auxiliary steam pipe, and from the exhaust of the evaporators and drain pipes, being led into it. The water may be passed through a layer of coke and sponge to cleanse it (Schultz system).

Remarks.—As it is necessary, for the preservation of the boiler, to prevent, as far as possible, air getting into it, care must be taken when arranging the feed pump suction pipes, that as little air as possible mixes with the water. If a filter is fitted in the feed pipe, a working pressure = boiler pressure + about 30 lb. per square inch must be allowed for. If it is fitted in the suction pipe or in the delivery pipe to the Weir's feed heater, no allowance need be made for any special pressure.

§ 258. **Feed-water Filter.**—Compare § 257 (6). Blake's filter may be cited as a typical example of this class of filter (Fig. 408). The water enters at the bottom of the hollow cylindrical space, and passes through filtering cloths, which are placed between alternate layers of

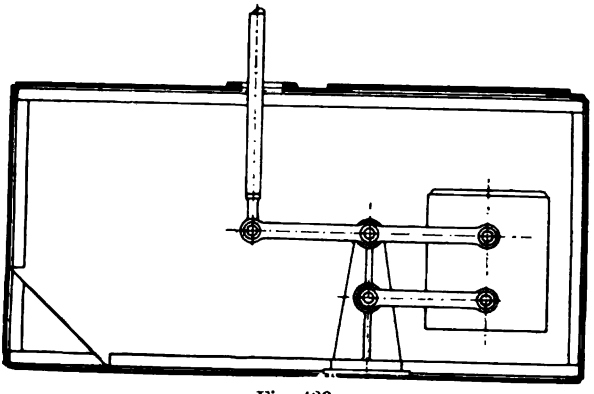


Fig. 409.

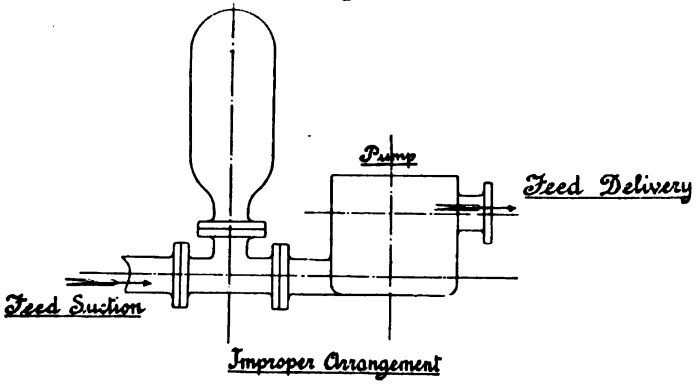


Fig. 410.

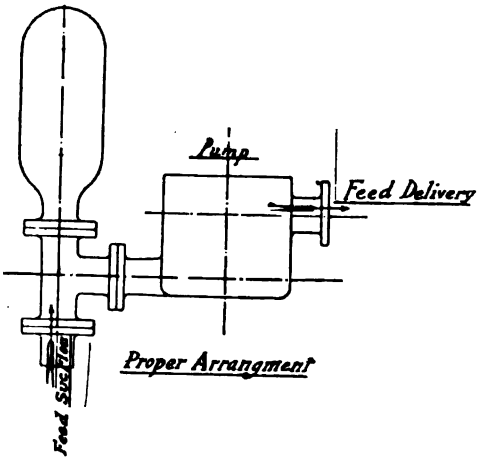


Fig. 411.

sieves, thirteen of which are shown one above the other in Fig. 408. The sieves and plates for directing the water are pressed together by a spindle. The clean water passes into the outer shell, and from thence into the pipe. To clean the filter the cloths are taken out, or else steam is forced through the reverse way, the dirt being washed down on to the plates, and then blown out through the central passage. A bye-pass and valves are fitted to the filter, so that the feed-water can be led direct to the boiler, while the filter is being cleaned.

§ 259. **The Float Tank** is generally a galvanised sheet-iron box (see Fig. 409) in which the float is arranged so that it is not disturbed by the intermittent action of the air pump-discharge. A rod carried upwards from the upper lever of the float works the cock supplying steam to the Weir feed pump. The float tank, besides a movable cover for admitting the float, and cleaning holes, has the following connections:—

Main feed pump suction.
Auxiliary feed pump suction.
Feed-heater pump suction.
Air-pump discharge.
Overflow from float tank.
Overflow from feed heater.
Auxiliary condenser air-pump discharge.
Jacket drain.
Cylinder drain.

§ 260. **Diameter of Suction and Delivery Pipes.**—Both the delivery and the long-suction pipes of the feed pumps should have air vessels, as near the pump as possible; these should be arranged so that the water enters from below. (See Figs. 410 and 411.) The suction pipes should be placed so that air pockets cannot occur, and should if possible be sloped up towards the pump. The diameters of the steam feed-pump, suction and delivery pipes should be so proportioned that the speed of the water in the former under normal working conditions is from 230 to 315 feet per minute, and in the latter from 300 to 500 feet per minute. The smaller values are for pipes where the ratio of length to diameter is large, the higher values for pipes where ratio of length to diameter is small. Using as a basis the quantities of feed-water per I.H.P. per hour given on page 310, the sectional area in square inches of the feed pipes per I.H.P. can be obtained from the following table:—

TABLE No. 52.

Sectional Area of Suction and Delivery Pipes of Feed Pumps per I.H.P. *for different Speeds of Water.*

Velocity of Water.	Compound Engines.	Triple Expansion Engines.	Quadruple Expansion Engines.
Feet per minute.	Square inch.	Square inch.	Square inch.
230	·00325	·00263	·00248
270	·00279	·00232	·00217
300	·00263	·00217	·00196
315	·00248	·00196	·00186
350	·00217	·00186	·00170
400	·00196	·00170	·00155
500	·00155	·00124	·00124

The total sectional area of the feed pipes in square inches may be found by multiplying the value in the table by the I.H.P. of the engine. The areas of the branch pipes leading off to the several boilers or groups of boilers must be of such dimensions that the velocity of the water through them is rather less than in the main feed pipe.

§ 261. **The Thicknesses of Copper Delivery Pipes** are determined in the same way as those of the steam pipes, but to allow for unavoidable shocks occurring in these pipes a pressure of 1·3 times the boiler pressure must be taken as the basis of calculation. Therefore—

$$\text{For drawn pipes } \delta = \frac{1{\cdot}3\ d\,.\,p}{5{,}688} + {\cdot}04 \text{ inch.}$$

$$\text{For brazed pipes } \delta = \frac{1{\cdot}3\ d\,.\,p}{5{,}688} \times {\cdot}06 \text{ inch.}$$

} See Table No. 53.

In war vessels the thicknesses are the same as for steam pipes, *i.e.*, they are calculated from the formula—

$$\delta = \frac{p \times d}{5{,}688} + {\cdot}06 \text{ inch.}$$

In the above formula d is the internal diameter of the pipe in inches, p boiler pressure in pounds per square inch above atmosphere.

TABLE No. 53.

Thickness of Copper Feed-water Pipes in Merchant Ships.

$$\delta = \frac{1{\cdot}3\ d\ .\ p}{5{,}688} + {\cdot}04 \text{ or } {\cdot}06 \text{ inch respectively.}$$

Inside Diameter.	130 lb. per square inch.		160 lb. per square inch.		185 lb. per square inch.		230 lb. per square inch.	
	Drawn.	Brazed.	Drawn.	Brazed.	Drawn.	Brazed.	Drawn.	Brazed.
Inches.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.	Inch.
1	0·068	0·088	0·068	0·088	0·078	0·098	0·088	0·108
1¼	0·078	0·098	0·088	0·108	0·098	0·118	0·108	0·127
1½	0·078	0·098	0·088	0·108	0·098	0·118	0·118	0·137
1¾	0·088	0·108	0·098	0·118	0·108	0·128	0·128	0·147
2	0·098	0·118	0·108	0·127	0·127	0·147	0·147	0·169
2¼	0·098	0·118	0·118	0·137	0·127	0·147	0·157	0·177
2½	0·108	0·127	0·118	0·137	0·137	0·157	0·169	0·187
2¾	0·108	0·127	0·127	0·147	0·147	0·169	0·177	0·196
3	0·118	0·137	0·137	0·157	0·169	0·187	0·196	0·216
3¼	0·127	0·147	0·147	0·169	0·169	0·187	0·206	0·226
3½	0·127	0·147	0·157	0·177	0·177	0·196	0·216	0·236
4	0·137	0·157	0·169	0·187	0·196	0·216	0·236	0·255
4½	0·157	0·177	0·196	0·216	0·226	0·246	0·265	0·285
5	0·177	0·196	0·206	0·226	0·236	0·255	0·285	0·305
5½	0·187	0·206	0·216	0·236	0·255	0·275	0·305	0·324
6	0·206	0·226	0·246	0·265	0·285	0·305	0·344	0·364
6½	0·216	0·236	0·255	0·275	0·305	0·324	0·364	0·383
7	0·226	0·246	0·265	0·285	0·314	0·334	0·383	0·403
7½	0·236	0·255	0·285	0·305	0·334	0·354	0·403	0·423
8	0·246	0·265	0·295	0·314	0·344	0·364	0·423	0·442

§ 262. **Feed Pipe Bends.**—Cast bends, tee-pieces, &c., for the feed pipes are, where economy has to be considered, made of cast iron; and in warships and other vessels, where strength and lightness are especially necessary, they are made of bronze. They must be so con-

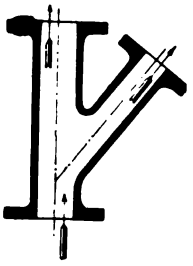


Fig. 412.

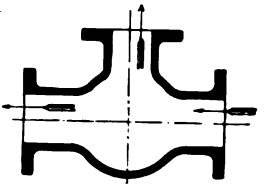


Fig. 413.

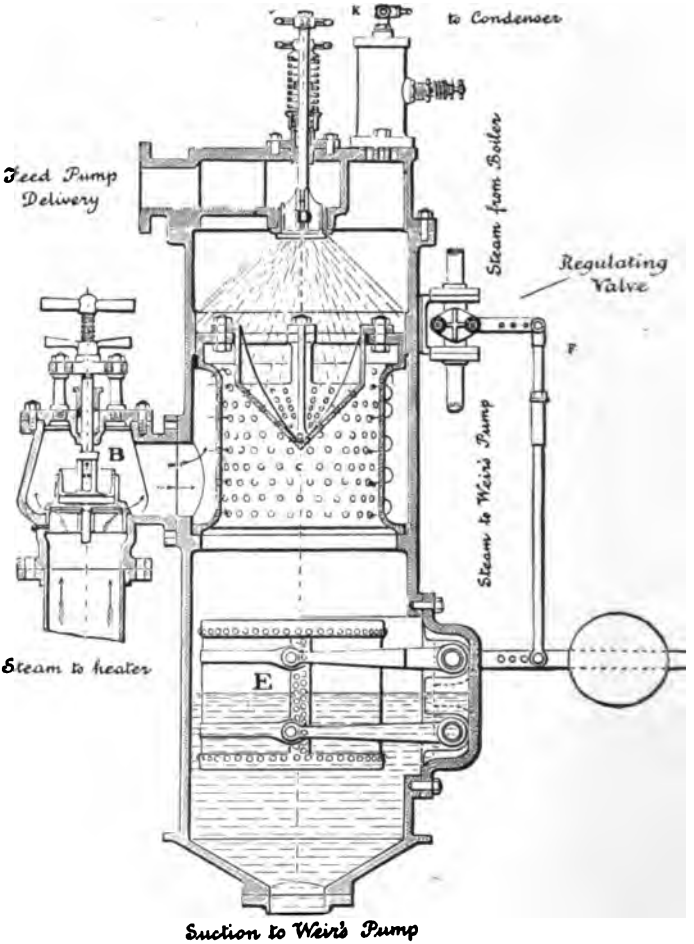


Fig. 415.

structed that the current of water in the branch pipes is disturbed as little as possible, and should be shaped as shown in Fig. 412. Globe tee-pieces (Fig. 413) are also sometimes used, but when fitted in pipes running in a straight line they cause a greater loss of pressure than pipes branching off at less than a right angle. Where there are several boilers, the pressure of water in the feed pipes in close proximity to the pump is naturally greater than in those further removed, but the effect of this can be minimised by the adoption of good arrangements of pipework, and by proper regulation of the feed valves.

§ 263. **Feed-water Heaters.**—A distinction is made between—

1. Surface feed heaters; and
2. Jet feed heaters.

1. In *surface feed heaters* the hot steam is either led through a system of tubes which are surrounded by the feed-water, or the reverse

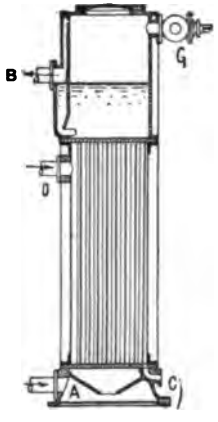


Fig. 416.

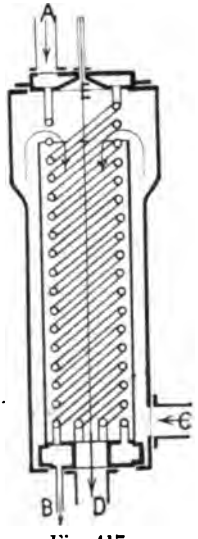


Fig. 417.

arrangement is adopted. The condensed steam is led either into the hot well or into the feed pump suction. The Lundkvist heater (Fig. 416) is fitted either into the air-pump discharge or the feed-pump suction. The feed-water enters at A, and is discharged at B; the hot steam (exhaust from auxiliary engines) enters at D, and the condensed water passes out at C. G is the blow-off cock.

The Pape & Henneberg heater (Fig. 417) is generally placed in the feed-pump delivery. The feed-water enters at C and passes out at D; the

live steam from the boiler enters at A, passes down through the coil of pipes, and is discharged as water at B. Air and oil may be blown off at E. The heaters may be cleaned by boiling out with soda.

2. Fig. 415 shows an ordinary Weir jet feed heater. The feed-water enters through valve D, and mixes with the exhaust from the auxiliary engines, &c., admitted through valve B. The heated feed-water collects in the lower part of the heater, and passes from thence to the steam pump, the speed of which is regulated by means of float E, which controls the cock F supplying steam to the pump. The liberated air can escape at K into the atmosphere, or be led off into the condenser.

Remarks.—There is no theoretical advantage in heating the feed-water, unless the exhaust from the auxiliary engines, evaporators, &c., is used for the purpose. If live steam is used to heat it, no economy from a theoretical point of view can be effected

SECTION V.

BILGE PIPES, BALLAST PIPES, CIRCULATING PIPES.

§ 264. **Bilge Pipes** are the pipes which serve to draw off the water collecting in different compartments of the ship, and to discharge it into the sea. They are generally made of lead, often of cast iron, but the small pipes connecting them with the *suction or mud boxes* are usually of copper. The connections at the ends of the different pipes are as a rule made with copper rings soldered on to the pipes, with loose wrought-iron flanges behind them. (See Fig. 383.) The thickness of the lead pipes (2 to 8 inches in diameter) is about $\delta = \frac{d}{12}$ to $\frac{d}{10}$.

The bilge pipes run under the floor-plates, and those passing through the cargo spaces must be protected from injury by strong wooden sleeves. They should not be carried through the coal bunkers, if it can possibly be avoided; but if no other arrangement is feasible, they should be strongly protected with wood. The lead or galvanised-iron strainers or mud boxes must be so placed that they can be easily examined and cleaned. They are frequently made in two halves, that they may be more conveniently taken asunder. The short piece of pipe attached to the suction-rose is made of galvanised iron.

Arrangement of Bilge Piping.—According to the regulations of the German Lloyd's, ships having engines of less than 100 I.H.P., and according to Lloyd's less than 70 I.H.P., must be provided with at least one bilge pump; ships with larger engines must have at least two such pumps, one of which is to be so arranged that it can draw from all the watertight compartments. An exception may be made in the case of the comparatively small watertight compartments in the extreme bows of the ship, for which it is only necessary to provide a small hand pump worked from the upper deck. Not only the engine bilge pump and the steam pump, but also the circulating pump, must be so arranged, that if necessary it can suck from the engine-room bilge through a non-return valve. The diameter of this suction pipe should be about 0·66 to 0·75 that of the circulating pump suction. In ships without a double bottom one suction amidships is sufficient in each compartment; but if the ship's bottom is fairly flat, it is desirable to have

a suction on each side of the ship, so that the bilge may still be pumped out if the ship should have a list to one side. There must of course be some arrangement for shutting off either of these suctions. Double-bottomed ships must have a bilge suction in each double bottom.

If there is a so-called "well," one suction amidships will suffice; but if the bottom of the ship is very flat, there must be one suction on each side. A separate suction pipe leads from each outlet to a common suction valve chest in the boiler or engine room, from which the bilge pumps draw. This valve chest should have non-return valves for each suction, so that if one watertight compartment is flooded, no water may find its way to another compartment through the bilge suction pipe. Each valve must be fitted with a name-plate, showing at a glance to which watertight compartment its suction pipe leads. The valves are often fitted so that they may be worked from the upper deck. The connection of the bilge suction valve chests to the bilge pump is generally through a three-way cock, the centre opening of which is connected to the suction pipe of the pump, the opening at one side to the bilge suction valve chest, and that at the other to a second valve chest, to which the suction pipes from the sea, ballast tank, boiler, condenser, &c., are connected. By this arrangement no water can pass from the sea, &c., to the bilge valve chests and thence to the interior of the ship. To prevent any dirt getting under the valves, cast-iron mud boxes or traps are generally fitted in each suction pipe, or between the suction valve chests and the pumps, in which the dirt is retained. These mud boxes must be easily accessible, and always above the floor-plates. The internal diameter of each bilge suction pipe should not be less than 2 inches in small ships, and as much as 4 inches in large ships.

§ 265. **Ballast Pipes.**—These serve to fill and empty the various ballast tanks. All the tanks of the double bottom, and also the extreme forward compartments, if they are used as ballast tanks, must be provided with at least one suction pipe leading to the ballast pump. For the extreme forward compartments, as well as for the tanks of the double bottom directly connected to each other, one suction pipe in the centre of each space is sufficient; but in larger ships, on account of the relatively flat bottom, it is necessary in the remaining tanks to have one suction pipe in the middle and one at each side, so that the tanks may be pumped dry when the ship has a list to either side. With this latter arrangement the suction pipes drawing from the middle of the ship are generally made larger than those drawing from the sides; the first is often called the *main ballast piping*, and the others the *auxiliary ballast piping*. The diameter of the latter is about 0·6 to 0·7 the diameter of the former.

These various pipes are united in suction valve chests in the engine and boiler rooms, which are connected by a single pipe to the adjacent ballast pump. The pipe from the valve chest to the pump may have a sectional area from two to three times greater than that of any of the separate suction pipes. Mud boxes are not necessary here, as in the case of the bilge suction pipes. The valves must be attached to their respective spindles, as they have to be kept open while the tanks are being filled. Each valve should have a name-plate showing where the suction pipe connected to it leads. The size of each suction pipe is generally so calculated, that, for a mean water speed of 200 to 350 feet per minute, the time required to empty all the tanks is from 4 to 5 hours, and from 2 to 2½ hours for a single tank. From this the size and duty of the ballast pump can be determined.

§ 266. **Diameter and Thickness of Ballast Pipes.**—The ballast pipes are generally of galvanised wrought iron or of cast iron.

Wrought-iron pipes are made in the following thicknesses :—

For internal diameter—

From 2 to 4 inches they are about 0·15 inch thick.
„ 4 „ 6 „ „ 0·17 to 0·20 inch thick.
„ 6 „ 8 „ „ 0·20 „ 0·23 „

The thickness of cast-iron pipes is about 0·4 to 0·5 inch.

The connecting pipes and bends and the bulkhead connections are generally of copper, about 0·12 to 0·15 inch thick. Care must be taken when fitting in the ballast pipes that the different branches have as few bends as possible, and that they are not carried through the coal bunkers, if it can be avoided. If they must pass through the bunkers or the hold, they should be protected from injury by strong removable wooden coverings. Sometimes these pipes are not run above, but through the double bottoms. This arrangement has the advantage that useful space in the bunkers and hold is not taken up by pipes and their coverings. On the other hand the pipes are not easily accessible. The ballast tanks are filled by means of a separate sea-valve, like that used for the circulating pump. The size of valve is proportional to the diameter of the ballast pump suction. As these pipes are generally of considerable length, it is best to have an air vessel near the pump, to avoid as far as possible shocks in the suction pipes.

The discharge pipe of the ballast pump is generally short, and is connected to a separate discharge valve on the side of the ship. This can usually be made to discharge above the water-line; and where this is possible, it is so fitted. A centrifugal or a plunger pump may be used as ballast pump. If the arrangements are such that the ballast pump is used for filling the tanks completely, a centrifugal pump

should be used, because if a plunger pump be employed, and the pumping is carelessly done, the ballast tanks might easily be burst or strained. If the ballast tanks are filled by means of plunger pumps, they must be provided with stand pipes, reaching to the upper deck, and open at the top.

§ 267. **Circulating Water Pipes.**—In engines of any considerable power, in order to prevent the bearings becoming overheated, should they tend to work hot, special pipes are provided for the proper circulation of cooling water. This cooling is effected by means of sea-water, applied either *directly*, by allowing the water to flow actually over the part of the engine affected, or *indirectly* by allowing the water to circulate through hollow spaces in contact with the surfaces to be cooled. Cranks, eccentrics, crossheads, plummer blocks, thrust blocks, stern tubes, &c., are "*directly cooled*"; while the crosshead guides are almost always, and the thrust collars, tunnel bearings, and shaft bearings are sometimes "*indirectly cooled.*" The water for the direct cooling is only applied when necessary, while that for the indirect cooling is always in circulation. The latter generally runs directly from the parts that have been cooled into a collecting pipe leading to the suction chamber of the circulating pump, sanitary pump, or some other sea-water suction chamber; the former passes into the bilge, and is pumped overboard by the bilge pump. The water is drawn from the delivery side of the circulating or sanitary pumps, or straight from the sea. The water circulation is conveyed in pipes of from $\frac{3}{4}$ to $\frac{1}{2}$ inch internal diameter. When the cooling water is directly applied, each pipe is so arranged that it can be separately shut off. Joints are provided so that water can be sprayed on to any part of the engine that is working too hot. Cocks and hose-connections are fixed in suitable positions, so that in case of need large volumes of water may be directed on to any part which requires to be rapidly cooled. In very large installations portions of the cooling water systems are sometimes kept separate, there being different pipes for instance for the thrust blocks and shaftings; or the direct and the indirect cooling water systems may be quite apart from one another.

The diameter of the cooling pipes for both direct and indirect cooling of the thrust bearings, shafting, and other parts of the main engine must be—

					Internal Diameter.
For engines from about	10,000	to	15,000	I.H.P.	$3\frac{1}{2}$ to $4\frac{1}{2}$ inches.
,, ,,	7,000	,,	10,000	,,	3 ,, $3\frac{1}{2}$,,
,, ,,	4,000	,,	7,000	,,	$2\frac{1}{2}$,, 3 ,,
,, ,,	1,000	,,	4,000	,,	2 ,, $2\frac{1}{2}$,,
,, up to about	1,000			,,	1 ,, 2 ,,

PART V.

STEAM BOILERS.

SECTION I.

FIRING AND THE GENERATION OF STEAM.

§ 268. **General Remarks.**—The function of a boiler is the conversion of the energy of a combustible into available heat. According to the Helmholtz law of the conservation of energy, the chemical composition of the coal determines the amount of heat which, under the most favourable theoretical conditions, can be obtained from it. The whole process of combustion is, however, so complicated, that the actual result attained varies greatly according to circumstances, and it is therefore necessary to study how it can be best carried on, the practical means to be adopted, and the effect which they may be expected to produce.

The subject divides itself naturally under two heads:—

(*a*) *Generation of heat on the grate. Process of combustion.*

(*b*) *Transmission of the heat obtained to the water in the boiler: Generation of steam.*

§ 269. **Process of Combustion.**—In general this may be regarded as the combination of a substance with the oxygen of the air, producing the phenomenon of fire. Chemical combinations are formed according to definite proportions of weight; therefore the quantity of air required for the combustion of a given quantity of combustible depends upon the composition of both (Table No. 54). A chemical combination is nearly always accompanied by the evolution of heat. It depends quantitively upon the character of the process (as shown by the products of combustion). Again, the quantity of heat obtained from the complete combustion of equal parts by weight of the same substances is always the same. It is thus possible to draw up tables from which we can calculate approximately the heat developed by the combustion of every known combustible (Table No. 54). From this table formulæ can be worked out for the quantity of air theoretically required, and the heat generated per unit weight of substance burnt (heating value). If a combustible is composed of certain percentages by weight of carbon (C), hydrogen (H), and oxygen (O), then the

Heating value $Q = 14{,}500\ (C + 4{\cdot}28H)$ thermal units per pound.

TABLE No. 54.

Complete Theoretical Combustion of the Constituents of Coal.

Chemical Composition.		Products of Combustion.		Pounds of Oxygen theoretically required per pound of Substance burnt.	Pounds of Air theoretically required per pound of Material.	Cubic feet of Air theoretically required at 62° Fahr. and 30 inches Mercury per pound.	Thermal Units generated per pound.	Theoretical Temperature of Combustion, Degrees Fahr.
Oxygen - -	O	—	—	—	—	—	—	—
Hydrogen -	H	Steam (water) -	H_2O	8·0	34·8	457	62,032	5,882
Carbon - -	C	Carbonic oxide	CO	1·33	5·8	76	4,452	2,372
„ - -	C	Carbonic acid	CO_2	2·66	11·6	152	14,500	4,982
Methane - (Light carburetted hydrogen.)	CH_4	Carbonic oxide and steam -	$CO + H_2O$	4·00	17·4	229	23,513	9,662
Ethylene - (Olefiant gas.)	C_2H_4	Carbonic acid + steam -	$CO_2 + H_2O$	3·43	15·0	196	21,343	9,752
Carbonic oxide	CO	Carbonic acid	CO_2	0·57	2·48	33	4,325	5,522
Sulphur - -	S	Sulphurous acid gas - -	SO_2	1·00	4·35	57	4,032	3,632

and the quantity of air by weight theoretically required to be added to it for complete combustion would be

$$L = 0{\cdot}116\ (C + 3H - 0{\cdot}4O)$$ lb. per pound, or by volume,

$$L = 1{\cdot}52\ (C + 3H - 0{\cdot}4O)$$ cubic feet per pound at 62° Fahr. and at 30 inches mercury.

More accurate values of Q can only be obtained from calorimetric determinations. (See Part VI.)

For summary of the heating value of various kinds of coal (mean of numerous measurements), see below, Table No. 55.

TABLE No. 55.

Heating Values of Various Kinds of Coal, &c.

Kind of Coal.	Heating Value, *i.e.*, British Thermal Units generated per pound when Combustion is complete.*
	B.T.U.
Welsh	14,850
Newcastle	14,800
Scotch	14,100
Westphalian hard coal	13,500 to 14,040
Silesian „	12,060 „ 12,420
Saxon „	10,980 „ 11,340
Saar „	12,960 „ 13,680
Illinois „	9,000 „ 10,800
Missouri „	9,900 „ 11,700
Pennsylvanian „	10,800 „ 13,320
Tasmanian „	about 10,800
Chilian „	„ 11,700
Japanese „	„ 12,600
Soft brown coal (lignite)	„ 7,200
Hard „	„ 9,900
Dry wood	3,600 to 5,400
Refined petroleum	18,000
Distilled petroleum residuum (Mazout)	18,000

§ 270. **Incomplete Combustion.**—In practice these values are

* By heat unit is meant the quantity of heat required to raise the temperature of 1 lb. of water 1 degree Fahr.

never attained, as combustion is always more or less incomplete. The degree of completeness achieved depends in the main on three factors :—

1. The quantity of oxygen introduced with the air for combustion.
2. The temperature in the combustion chamber.
3. The intimate mixing of the fuel and oxygen at the place where combustion takes place.

To appreciate more fully the effect of these factors, it will be useful to illustrate the process of combustion by citing a typical example. For this purpose let us select the ordinary cylindrical boiler with a horizontal grate, furnace, fire box, and boiler tubes ; there will be no difficulty in drawing deductions for other types of boilers.

The process of combustion is generally as follows :—The air for combustion, entering below the grate through the ash-pit, meets the lowest layers of coal, which are at a white heat, and carbonic acid gas is generated. Complete combustion takes place here, because a large excess of air is always present. The higher the mixture of nitrogen, carbonic acid gas, and oxygen, rises through the coal on the grate, the less becomes the excess of air. Combustion is retarded, in consequence of the fall in temperature produced as the upper layers of coal are reached. A mixture of gaseous hydrocarbons given off from the fresh coal (such as methane, ethylene, see Table No. 54), carbonic oxide, carbonic acid, oxygen, nitrogen, &c., is therefore formed, and appears as flame during combustion. We have thus to distinguish between two stages of combustion, viz., that of the solid carbon in the lower strata, and that of the gaseous constituents above the layer of fuel. The poorer or the richer a coal is in hydrocarbons, the more or the less important will be the second stage, with regard to the total process of combustion, and especially with regard to the production of smoke. The opinion is widely entertained that smoke consists of small particles of unconsumed carbon carried off from the grate. But a close examination of the nature of the soot in the smoke shows that soot is a residual product of combustion itself, and is present in largest quantities where the coal consumed is rich in hydrocarbons.

§ 271. **Losses by Excess of Air.**—The theoretical quantities of air given in § 269 as requisite for the complete combustion of the coal are not sufficient in practice, because of the great difference in volume between the air and the coal (with the theoretical quantity of air this ratio is about 12,000 to 1), and also because of the impossibility of exercising a rigid control over the movement of the air through the fire. Combustion must therefore be carried out with an excess of air, but this should be restricted within the narrowest limits, because the

heat carried off uselessly up the chimney increases with the quantity of gases of combustion :—these escape at a temperature of about 570° Fahr., and thus the heat efficiency of the boiler is lowered. (Compare Table No. 56.) In practice, when calculating the dimensions of the grate and flues, an excess of air of 100 °/₀ is generally allowed for, and the working of the fire is regulated by dampers and ashpit doors.

TABLE No. 56.

Heat Losses in the Gases of Combustion.

(The loss is expressed in percentages of the heat generated on the grate.)

Temperature of Gases of Combustion.	Percentage of Excess of Air.						
	0	25	50	75	100	125	150
Deg. Fahr.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
480	8	10	12	14	16	18	20
570	10	12·5	15	17·5	20	22·5	25
660	11·5	14·4	17·3	20·2	23·1	26	28·9
750	14·2	17·8	21·3	24·9	28·4	32	35·5

The above figures apply only on the assumption that pure carbon is burnt on the grate. In practice the losses resulting from too great an excess of air are higher.

§ 272. **Grate Area.**—This is the starting point for designing boilers, as a square foot of grate area is taken as a basis of all calculations. To determine the actual dimensions of the total grate area for a given boiler or set of boilers, it is necessary to know from experimental data the *horse-power generated per square foot of heating surface, and the ratio of heating surface to grate area.* As the dimensions of *each* grate are determined by convenience in stoking, the number of grates required is easily arrived at. According to the *quantity of coal burnt, whether more or less, per square foot of grate area,* combustion is said to be either normal or more or less forced. The values given in Table No. 57 are the result of extensive practical experience. For more precise values, for different types of boilers, see Tables Nos. 63 to 67 ; for cylindrical boilers, see Table No. 62.

TABLE No. 57.

Pounds of Coal burnt per square foot of Grate, Quantity of Air required, and Pressure of Air or Vacuum for the three Methods of Firing in most general use.

Draught.	Coal burnt per square foot of Grate Area, pounds per hour.	Quantity of Air required per square foot Grate per hour, taking 100 % Excess of Air as a Basis.		Pressure or Vacuum in inches of Water.
		Pounds per hour per square foot Grate.	Cubic feet per square foot Grate per hour at 62° F. and at 30 inches Mercury.	
Natural draught - -	15 to 25	368 to 543	4,900 to 7,200	0·19 to 0·50
Induced draught - -	22 ,, 30	543 ,, 717	7,200 ,, 9,500	0·39 ,, 1·58
Forced draught { Moderate -	22 ,, 37	543 ,, 880	7,200 ,, 11,800	0·39 ,, 0·79
Forced draught { High - -	37 ,, 50	880 ,, 1,230	11,800 ,, 16,400	0·79 ,, 2·75

Remarks.—1. The inches of water give the difference in pressure between the ash-pit and the external air, if they show pressure, and between the external air and the combustion chamber, if there is a vacuum (induced draught).

2. In torpedo-boats and destroyers, at full speed, the consumption of fuel may reach as much as 80 lb. per square foot of grate per hour.

The choice of the system of firing to be adopted, whether *natural* or *artificial* (induced and forced draught), is determined by the desired rate of combustion of coal on the grate, or evaporation in the boiler. For slow-going freight and passenger steamers natural draught is preferable, because simpler. In fast steamers all three kinds are in use, but only with moderate rates of forcing. War vessels employ almost exclusively forced draught, and the coal burnt per square foot of grate varies according to circumstances.

§ 273. **Natural Draught** is produced by the difference in specific weight between the hot gases of combustion passing up the chimney and the outer air. The amount of excess of pressure of the outer air below the grate is determined by the height of the funnel above the grate, mean temperature of the escaping gases of combustion, and their speed through the flues, and the temperature of the external air. Let

h be the head produced by the chimney draught at the grate, in inches of water;

H be the height of the funnel above the grate in feet;
t_2 „ mean temperature of the chimney gases, degrees Fahr.;
t_1 „ temperature of the outer air, degrees Fahr.; then

$$h = \frac{H}{66{\cdot}6}\left(1 - \frac{t_1+461}{t_2+461}\right) - {\cdot}12 \text{ inch.}$$

The degree of pressure required to maintain the speed of the gases against the resistance in the flues is brought into the above formula by means of the constant ·12 to be subtracted, which corresponds to a mean value, assuming the sectional area of the flues to agree with those given in Table No. 58.

TABLE No. 58.

Mean Proportional Dimensions for Calculating the Sectional Area of Flues of Ship's Boilers with any kind of Draught.

The following sectional areas per square foot of grate may be used:—

In the ash-pit	·15 to ·20	square feet.
Over the fire bridge	·15 to ·18	„
In the vertical portion of the fire box	·30 to ·40	square feet (as large as constructive conditions will allow).
Through the tubes	·16 to ·20	square feet.
In the smoke box	·40 to ·50	„
In the uptake	·16 to ·25	„
In the funnel	·16 to ·24	„

N.B.—The cylindrical boiler has been taken as the basis for this table; for boilers of other types the required sectional area can be determined without difficulty, by substituting corresponding values, the lower values for natural draught, the higher for forced draught with heated air. In calculating the sectional area for the passage of the flue gases through the nest of tubes in water-tube boilers, the area allowed is about ·15 to ·2 square foot for every square foot of grate surface.

Example.—Cargo boat in northern latitudes—

$$H = 82 \text{ feet}; \; t_1 = 68^\circ \text{ Fahr. (max.)}; \; t_2 = 392^\circ \text{ Fahr.}$$

$$h = \frac{82}{66{\cdot}6}\left(1 - \frac{529}{853}\right) - {\cdot}12 \text{ inch} = 0{\cdot}47 \text{ inch} - 0{\cdot}12 \text{ inch} = 0{\cdot}35 \text{ inch.}$$

In this set of boilers the maximum amount of coal that could be burnt was 18·5 lb. per square foot of grate per hour. Assuming that if

the combustion were more forced, the gases of combustion would escape at a higher temperature, say a mean of 572° Fahr., then

$$h = \frac{82}{66{\cdot}6}\left(1 - \frac{529}{1{,}033}\right) - 0{\cdot}12 = 0{\cdot}48 \text{ inch},$$

so that the maximum quantity of coal burnt per square foot of grate area per hour might amount to 22·5 lb. If the same steamer were running in the Tropics, the temperature of the air might be 104° Fahr., and the mean exit temperature of the gases of combustion, 392° or 572° Fahr., then h would be

$$\frac{82}{66{\cdot}6}\left(1 - \frac{565}{853}\right) - 0{\cdot}12 \text{ inch} = 0{\cdot}3 \text{ inch, or}$$

$$\frac{82}{66{\cdot}6}\left(1 - \frac{565}{1{,}033}\right) - 0{\cdot}12 \text{ inch} = 0{\cdot}42 \text{ inch}.$$

The difference is comparatively small.

§ 274. **Artificial Draught.**—If the quantity of coal burnt per square foot of grate area per hour exceeds 20 lb., the quantity of air obtained with natural draught is frequently insufficient, and a *supplementary supply* must be *artificially provided.* According to the method of admitting this air, whether it is drawn into or forced through the fire, the system is known as induced draught, or forced draught. The principal types of the different systems are :—(1) *Induced draught*—Ellis and Eaves; (2) *Forced draught* (*a*) with closed stokehold, (*b*) with closed ashpit—Howden and other systems. For the usual arrangement of these systems of firing, see Figs. 481 to 489. The dimensions of the sectional areas of the various flues given in Table No. 58 can be used provided that the higher values therein stated be taken.

§ 275. **Centrifugal Fans.**—The most important adjuncts to these forced draught systems are the *centrifugal fans*, which are almost exclusively used in ships to furnish the supply of air required. For calculating the dimensions of the fans, the quantity of air to be delivered is taken as a basis, as also the pressures of air (or vacuum) required, and the power necessary to produce them. It should be noted that the pressure produced by the fan must be added to that due to the natural draught.

§ 276. **The Dimensions of the Fans** may be determined by means of the subjoined calculations where the following factors are assumed :—The overall diameter of the vane wheels of a fan is from one and a half times to twice the internal diameter.* The circum-

* In practice the values for the external diameters are $r_a = 2$ feet 6 inches to 4 feet for forced draught ; $r_a = 3$ to 5 feet for induced draught.

ferential sectional area at any point of the fan measured parallel to the axis, and at any distance from it, is made the same as the cross section of the intake, and kept a constant throughout the fan, and according to its size it is from 0·6 to 0·75 the sectional area of the flue in which the fan is to be placed. Therefore in proportion to the grate area, the radial cross section would be (see Table No. 58):—

1. For *induced draught*, from ·12 to ·185 square feet per square foot of grate area.

2. For *forced draught*, from ·09 to ·12 square foot per square foot of grate area.

By the use of these figures the velocity of the air through the fan may be determined for every given case.

§ 277. **Example I.**—Boiler with *closed stokehold.* Given a minimum of 22·5 lb. and a maximum of 37 lb. of coal per square foot of grate area per hour. According to Table No. 57 we shall therefore require per hour from 7,200 to 11,800 cubic feet of air at 62° Fahr. and 30 inches mercury. The cross section of the fan (which in this case is supposed to deliver direct to the ash-pit) we will take according to the above data at 0·1 square foot per square foot of grate area. We thus get a velocity of air through the fan of

$\frac{7,200}{60 \times 60 \times 0{\cdot}1} = 20$ feet per second, and $\frac{11,800}{60 \times 60 \times 0{\cdot}1} = 32{\cdot}8$ feet per second.

If the system of draught under consideration were induced draught, it would be necessary to note that, on account of the higher temperature of the gases, the volume of air and also its speed would increase in direct proportion to the absolute temperatures. (Compare also Table No. 58.)

§ 278. **Example II.**—Given a boiler with *induced draught* burning 28·7 lb. of coal per square foot of grate area per hour. According to the above rule the cross section of the fan wheels is 0·15 square foot per square foot of grate area. The quantity of air required at 62° Fahr. and 30 inches mercury is, according to Table No. 57, 9,000 cubic feet per square foot of grate area per hour. At 752° Fahr. that is

$$9,000 \times \frac{752+461}{62+461} = \text{about } 21,000 \text{ cubic feet per square foot grate.}$$

(The reduction in pressure may be neglected as the calculation is based on the volume.) Thus the radial velocity of the gases through the fan will be $\frac{21,000}{60 \times 60} \times \frac{1}{0{\cdot}15} = (\text{about})\ 39$ feet per second.

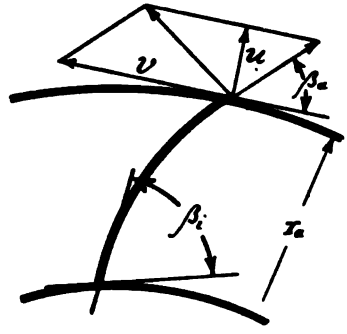


Fig. 418.

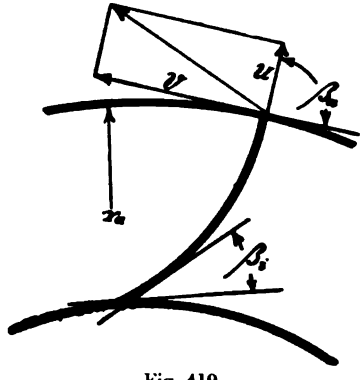


Fig. 419.

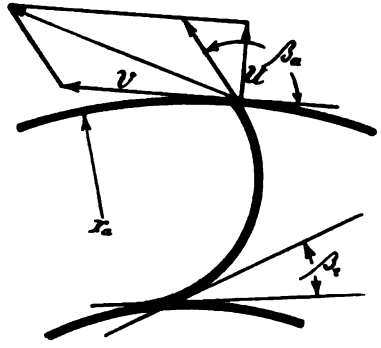


Fig. 420.

§ 279. **The Form of the Vanes** (or blades) to ensure the delivery of a given quantity of air is of great importance, because it determines the number of revolutions, the air pressure obtainable, the maximum quantity of air that can be delivered with a given breadth of vane, and the work expended to produce it. As the fans in every system are called upon to run at very varying speeds, an entrance of the air free from churning at all speeds cannot be ensured. From practical considerations the angle β_1 (Figs. 418 to 420) should not be taken too small, *i.e.*, not less than 30°. The *external angle of the blade* may vary from 60° to 120°.

The actual *shape of the blade* is preferably designed with arcs of circles or straight lines, care being taken that no sudden changes of direction occur. The shape having been determined, the pressure to be attained with a given number of revolutions, and the expenditure of work required to produce it, are also determined, the breadth of the blade being ascertained by assuming a given velocity of air. By pressure is to be understood the difference in inches of water between the delivery and suction sides of the fan, it being a matter of indifference whether the latter forces the air through or sucks it in. The difference in the weight of the air in either case is too small to appreciably affect the results. Let

r_a denote the radius of the fan disc in feet.

n denote the number of revolutions of the fan per minute.

h_1 „ difference in pressure between the suction and delivery sides in inches of water.

u „ radial velocity of the air through the fan (a constant varying with the type of construction), in feet per second.

v „ velocity of the fan disc at its outer periphery in feet per second.

β_a „ external blade angle. (See Figs. 418 to 420.)

T „ absolute temperature in degrees Fahr. of the air passing through the fan.

f „ circumferential sectional area in square feet of the fan disc per square foot of grate area measured concentrically to the axis, and at right angles to the radius (see § 276), constant throughout the disc.

A „ work in H.P. expended in driving the fan per square foot of grate area.

$$\text{Then } h_1 = \frac{\cdot 1}{T}(v - u \cot \beta_a)^2.$$

$$A = \frac{\cdot 02 fu}{T} \times [\cdot 093(v - u \cot \beta_a)^2 + (\cdot 305u)^2].$$

Example I.—Boiler with *induced draught* but without air being previously heated. Total grate area 484 square feet; 25·6 lb. of coal to be burnt per square foot of grate area per hour. From Table No. 57 there are required per square foot of grate area per hour, 8,500 cubic feet of air at 62° Fahr. As the fans are placed in the uptake, the quantity of air at 752° Fahr. to be delivered per hour is—

$$8{,}500 \frac{752 + 461}{62 + 461} = \text{about } 20{,}000 \text{ cubic feet per square foot grate.}$$

The cross section (f) according to the data on page 461 is 0·15 square foot per square foot of grate. Then—

$$u = \frac{20{,}000}{60 \times 60 \times 0{\cdot}15} = 37 \text{ feet per second.}$$

To keep the engines and fans as light as possible, a high value for v about = 82 feet per second is taken, always keeping in view the strength of the material.*

* For parts of machines rotating at high speeds, such as, for instance, the circumference of fly-wheels, a simple calculation will prove that the stress upon the material due to centrifugal force depends entirely on the circumferential velocity, and is not affected by the radius or the number of revolutions.

Then according to the above formula, with $\beta_2 = 90°$

$$h_1 = \frac{\cdot 1}{752 + 461} \times (82)^2 = \text{about } 0{\cdot}55 \text{ inch.}$$

This is the difference in pressure between the suction and delivery of the fan. To this must be added the vacuum produced by the natural draught (see page 458), so that the difference in pressure between the suction side of the fan and the outer air is about 0·55 inch + 0·35 inch = 0·9 inch. Above the fires the vacuum will naturally be somewhat less, owing to the resistance offered by the flues between the combustion chamber and the fan to the passage of the air through them. The work required will be—

$$\text{A} = \frac{\cdot 02 \times \cdot 15 \times 37}{752 + 461}[\cdot 093(82)^2 + (\cdot 305 \times 37)^2] = \cdot 095 \text{ H.P.}$$

per square foot of grate area. Total expenditure of work for 484 square feet of grate area = 46·2 H.P. This is the work required at the shaft of the fan. Suppose the diameter of the fan be taken (according to the space available) at 8 feet. Then—

$$n = \frac{60 \times 82}{8 \times \pi} = 195 \text{ revolutions per minute.}$$

The number of fans is generally determined by the grouping of the boilers and arrangement of the uptakes. Assuming that 4 were to be used, the external width per fan would be—

$$b = \frac{0{\cdot}15 \times 484}{4 \times 8 \times \pi} = 0{\cdot}73 \text{ foot} = \text{about } 8\tfrac{3}{4} \text{ inches.}$$

With these measurements the fan must be constructed so that it is not contracted in any part, or if so it must be made correspondingly wider. If fans of these dimensions were used with a boiler fitted with the Ellis and Eaves system of forced draught, in which the combustion gases are cooled by passing through an air-heater before they enter the fan, T would be 853° instead of 1,213° Fahr. absolute, and

$$h_1 = \frac{0{\cdot}1}{853} \times 82^2 = 0{\cdot}78 \text{ inch.}$$

Total vacuum = 0·78 inch + 0·35 inch = 1·13 inch.

Then $u = 25{\cdot}7$ feet per second, and

$$\text{A} = \frac{\cdot 02 \times \cdot 15 \times 25{\cdot}7}{853}[\cdot 093(82)^2 + (\cdot 305 \times 25{\cdot}7)^2] = \cdot 075 \text{ H.P.}$$

Total work done on the fan shaft, $484 \times \cdot 075 = 36{\cdot}2$ H.P.

Example II.—Boiler with *closed stokehold*; total grate area, 970 square feet; maximum quantity of coal to be burnt per square foot of grate area per hour, 35·8 lb. According to Table No. 57, the quantity of air required per square foot of grate area per hour is 11,500 cubic feet. Cross section, from page 461, 0·1 square foot per square foot of grate area.

$$u = \frac{11{,}500}{0{\cdot}1 \times 60 \times 60} = 32 \text{ feet per second.}$$

Assuming v, as before, at 82 feet per second, $\beta_a = 90°$, and T at (68° Fahr. + 461). Then—

$$h_1 = \frac{0{\cdot}1}{529} \times 82^2 = 1{\cdot}26 \text{ inch.}$$

Total excess of pressure between the stokehold and combustion chamber = 1·26 inch + 0·35 inch = 1·61 inch.

$$\text{A} = \frac{{\cdot}02 \times 0{\cdot}1 \times 32}{529}[{\cdot}093(82)^2 + ({\cdot}305 \times 32)^2]$$

= about ·113 H.P. per square foot grate.

Total work expended on the fans is therefore about 110 H.P.

There are assumed to be five stokeholds, each having four fans.

Taking the diameter of the fans as 5·9 feet (see page 460), then

$$n = \frac{60 \times 82}{5{\cdot}9 \times \pi} = 265 \text{ revolutions per minute.}$$

The circumferential breadth should be $b = \frac{970 \times 0{\cdot}1}{20 \times 5{\cdot}9 \times \pi} = 0{\cdot}262$ foot, if there are no contractions or losses from other sources. If any such have to be provided for, the fans must be made correspondingly wider, and the power of the fan engine increased. These contractions may occur either in the fan or in the flues themselves. The most helpful rule in practice is to *calculate the fans* and engines for a delivery of about *double the quantity* of air actually required, and to adjust the right number of revolutions by the stop valve, and the cross section of the air passages by suitable dampers while the fan is at work.

If the fans are driven, not by steam engines, but by electric motors, which must be run at a given number of revolutions per minute, the output of the fan must be regulated by the dampers. For this class of motor, fans with bent-back blades ($\beta_a < 90°$) are not suitable, because if the quantity of air increases, the pressure diminishes when the speed remains constant. Fans should be used with blades bent somewhat forward ($\beta_a > 90°$), in which the pressure increases with the quantity of air admitted, when the revolutions remain constant. With blades which are radial throughout their length the pressure is constant for

any given quantity of air. The most important source of loss of air with closed stokeholds is the leakage of air through openings and cracks. This question, however, cannot be treated here; it is best to avoid these causes of loss by care in details of construction.

§ 280. **The Number of Blades** varies, according to practical conditions, from ten to sixteen, but frequently only half of them are carried out from r_i to r_a (Fig. 420), while the other half are fitted only from the outer periphery down to half the radial depth of the wheel. The outer angle of the blade β_a has a great effect upon the number of revolutions.

As for the same h_1,

$$(v - u \cos \beta_1) = \text{constant, and must be} = \text{about } v_1,$$

then for example—

$$\begin{aligned} \text{With } \beta &= 60^\circ & v &= v_1 + 0{\cdot}577u \\ \text{,, } &= 90^\circ & v &= v_1 \\ \text{,, } &= 120^\circ & v &= v_1 - 0{\cdot}577u \end{aligned}$$

Therefore in our example where $v_1 = 82$ and $u = 37$ (feet per second). with $\beta = 60^\circ$ and 120° respectively, the corresponding values would be 61·4 feet and 102·6 feet per second, which would cause a considerable difference in the number of revolutions.

§ 281. **A High Temperature in the Combustion Chamber** is an essential condition for obtaining complete combustion, because the ignition of the combustible constituents of the coal requires certain definite temperatures, and the time available for the process of combustion is very short. As the air on entering must first be heated before it is in a position to assist combustion, the intensity of combustion depends on the excess of air. It is also clear that in any case the temperature of combustion to be attained depends directly on the excess of air, because the whole of the heat generated by combustion passes first into the products of combustion, which in their turn convey it by radiation to the surrounding substances. Therefore the admission of only the theoretical quantity of air required for complete combustion theoretically yields the highest temperatures of combustion. (Compare Table No. 54.)

In practice, having regard to the material of the furnace, &c., excessively high temperatures in the combustion chamber should be avoided; nor are they practically attainable on account of the excess of air required for other purposes (see § 271). Nevertheless, when working the furnaces, great care must be taken that (by bare patches occurring in the grate, or by the fire doors being left open too long, &c.) the excess of air does not become so considerable that the flame is extinguished by

lowering its temperature too far. In furnaces with artificial or forced draught, in which the speed of the air through the grate is comparatively high, it is desirable, in order to assist complete combustion, to warm the air beforehand, or to inject heated air into the space above the grate (secondary admission of air for combustion, as used by Howden, Belleville, and others).

§ 282. **Mixing of the Gases of Combustion.**—The complete and thorough *mixing* of the oxygen of the air and the combustible constituents of the coal is very important in order properly to carry out combustion. In its first stage (see § 270) this mixing is easily effected, because the coal remains stationary, and the air passes over it; but in the second stage it is much more difficult to effect, because the constituents are now gaseous, at a relatively high temperature, and therefore having a high specific volume, move at equal velocity side by side through the flues. It is only possible to mix them if there are frequent changes of direction, and if eddies are formed, as for instance at the fire bridges, or by the injection of secondary air or steam over the grate.

§ 283. **The Useful Heat of Combustion** is that heat contained in the hot gases which can be transferred to the water in the boiler. The losses here are as follows:—Heat radiated externally from the furnace, heat remaining in the ashes, incomplete combustion, and heat carried off in the chimney gases. External radiation from the furnace may be neglected, as the air surrounding the boiler, which is afterwards drawn into the furnace, absorbs the greater part of it. Losses in the burnt residuum (ashes, &c.) seldom amount to more than 3 °/₀, if the grate be in good condition and properly stoked. Losses arising from incomplete combustion may, under certain circumstances, be very considerable. The cause of them in furnaces of ships' boilers is not the want of air for combustion, for there is always a large excess of air, but from the too great reduction of the temperature of the flame due to keeping the fire doors open too long, from putting on too large a quantity of coal at once and in lumps of too large a size, from allowing the grate to become clogged, &c.* If the fires be skilfully stoked, these losses may be kept within comparatively narrow limits, so that the principal loss in the whole process consists in the heat carried off by the gases. The amount of this loss depends wholly on the excess of air, and may under certain con-

* In ships' boilers a third of the fires are cleaned at the beginning of each watch (*i.e.*, every four hours). Therefore at the beginning of each watch the evaporation diminishes in a marked degree, and the speed of the engines may be reduced from 3 to 5 %.

ditions be considerable. (See Table No. 56.) Experiments to determine it in any existing boiler can, in ships, only be carried out by chemical analysis of the flue gases (see Part VI.), that is, by ascertaining the excess of air from the percentage of oxygen contained in the exhaust gases.

§ 284. **Generation of Steam.**—The transmission of heat from the gases of combustion to the contents of the boiler takes place by conduction. To effect this transmission economically through the dividing walls, the latter should be as thin as possible, and their surface as great as possible (heating surface). Strength and tightness are the first practical considerations to be attained. The endeavour to have walls as thin as possible and yet sufficiently solid has led to the construction of water-tube boilers. The *amount of heating surface* is determined by the final permissible temperature of the hot gases, having regard to the utmost utilisation of the heat in them.

Purely theoretical considerations are of little value here. In the course of years of practical experience certain data have been obtained for different types of boilers, which are classified in Table No. 59. With the same kind of flues and the same draught, merchant vessels show higher values of $\frac{\text{heating surface}}{\text{grate surface}} = \frac{\text{H.S.}}{\text{G.S.}} = \frac{\text{H}}{\text{R}}$ than warships, as in the latter more importance is attached to utilising the weight of metal to the utmost, and no attempt is made to turn to best account the heat contained in the gases of combustion when the engines are working at full speed.

§ 285. **The Efficient Transmission of Heat from the Gases of Combustion to the Water** requires the closest contact between the gases and the walls of the boiler on the one hand, and between the water and the boiler wall on the other. To obtain this it is necessary that the heating surfaces should be kept clean inside and out (from soot, ashes, salt, or other deposits). The process of transmission of heat is also greatly assisted by sudden changes in direction and of sectional area of the flues, and also by suitable arrangement of the fire bridges, &c.

§ 286. **The Heat Transmitted to the Contents of the Boiler** has first to heat up and then to evaporate the water. The theoretical quantity of heat available may be ascertained from Table No. XXII.

The values given in the different columns of Table No. XXII. (page 694) are obtained from the theory of the mechanical equivalent of heat, and require some explanation.

If water is heated in a closed space at constant pressure its temperature rises to boiling point, and *the amount of heat required to raise* 1 *lb. water through* 1° *Fahr. is a unit of heat or thermal unit.* The heat

TABLE No. 59.—*Ratios of Heating Surface to Grate Surface, and Evaporation in Ships' Boilers—Mean Values.*

Type of Vessel.	Kind of Draught.	Type of Boiler.	Ratio of Grate Area to Heating Surface.	Water Evaporated.			I.H.P. per square foot of Heating Surface.	I.H.P. per square foot of Grate Area.
				Per hour per square foot of Grate.	Per hour per square foot of Heating Surface.	Per pound of Coal.		
				lb.	lb.	lb.		
Cargo and passenger steamers	Natural	Cylindrical	1:25—1:35	140—205	5·7—5·8	9—9·5*	0·26—0·3	7·4—10·2
Fast steamers -	Natural	Cylindrical	1:30—1:35	185—225	6·1—6·4	9—9·5	0·28—0·33	10—13
	Artificial / Slightly forced		1:36—1:40	185—290	6·1—7·1	8·5—9	0·32—0·46	13—16·7
Ironclads and heavy cruisers	Forced	Cylindrical	1:28—1:32	245—325	8·2—10·9	8 max.	0·32—0·56	14—15·8
		Water tube	1:50			7·5 min.	0·37—0·46	14—20·4
Light cruisers -	Forced	Water tube	1:50	305—370	8·7—9·2	7·8 max.	0·51	
						7·25 min.	0·31—0·44	15·8—22·3
Torpedo, &c., boats	Forced	Water tube	1:50—1:60	up to 600	up to 10·25	6·5—7·5	0·46—0·56	21—25
		Locomotive						24—30

* These high values are only possible when so-called "retarders" are used, *i.e.*, spiral iron coils which are inserted in the boiler tubes, and have from one to two turns, according to the length of the tube.

required per pound to heat the water from 32° Fahr. to boiling point is sometimes called the *water heat of steam* (see column 3, Table No. XXII.). While boiling in an open vessel the temperature remains constant (temperature of boiling water). The steam generated is called *saturated* as long as water is present. At the moment when the last drop of water disappears the steam is *dry saturated.* According to Regnault's experiments the temperature of saturated steam is a function of the pressure only, and therefore in Table No. XXII. a definite temperature is given for each definite pressure.

The heat required to convert 1 lb. of water into steam at the same temperature is called *the latent heat of evaporation or total latent heat* (column 4 + column 5). This consists of the so-called internal latent heat and the external latent heat. The internal latent heat is used to overcome the resistance of the molecular forces which oppose the change of state of the molecules brought about by the formation of steam, while the external latent heat represents the amount of heat which has to be added to overcome the external pressure in pounds per square inch as the volume (of water or steam) increases. Thus the so-called total heat of 1 lb. of dry saturated steam consists of the water heat (column 3) and the latent heat (column 4 + 5). It signifies the excess of heat possessed by 1 lb. of dry saturated steam over 1 lb. water at 32° Fahr. The characteristic of saturated steam is that if heat be withdrawn from it, it does not produce a fall in temperature, but condensation. If dry saturated steam is still further heated, the pressure remaining constant, it becomes *superheated; i.e.*, it attains a condition in which it is more like air, because if heat be withdrawn from it, not condensation, but only a fall in temperature of the steam, results.

§ 287. **The Formation of Steam in the Water** produces violent ebullition, due to the sudden increase in volume of the steam generated. The eddies produced, and the escape of the steam from the surface of the water in the shape of bubbles, are the reasons for the formation of wet steam over the surface of the water. The water mechanically mixed with the steam passes through the engine, but performs hardly any useful work, and may easily give rise to water-hammer; the generation of dry steam should therefore be aimed at. The means to ensure this, apart from any fittings especially provided to dry the steam, are to provide large steam spaces above the water level, to prevent violent ebullition at the water surface by providing a large surface of water for disengaging steam, and the use of pure water, the distribution of the heat as evenly as possible over the total heating surface, together with a proper circulation of water, and lastly a moderate evaporation per square foot of heating surface (no undue forcing should be allowed).

The dimensions of the steam space and the level of the surface of the water are almost always fixed in ordinary cylindrical boilers by their construction. In water-tube boilers both steam and water spaces are generally much reduced, because the adoption of these boilers is dictated by the urgent desire to economise weight, and therefore they yield much wetter steam than cylindrical boilers. As it is all important to avoid scum in marine boilers in which the same water is used over and over again, special fittings are necessary to remove all oil or greasy deposit from the feed-water. To generate the steam efficiently the heat should be transmitted as uniformly as possible over the whole heating surface. It is a well-known fact that uniform transmission of heat forms an important factor in the efficient generation of steam. (For circulating apparatus see § 333.) It is necessary that the water evaporated from the heating surface should be continuously replaced, and the best means of effecting this is to ensure, by the aid of some auxiliary arrangement, a regular flow of water to the heating surface. No certain data on this point can, however, be obtained, as opinions as to the value of the different methods vary greatly. (See *Engineering*, 1896, page 583.) In water-tube boilers, which have headers to convey the water fed into the upper drum down to the lower, the total sectional area of the headers is calculated to give ·032 square inch per square foot of heating surface. In cylindrical boilers the water rises in the neighbourhood of the combustion chamber, and descends towards the front tube plate.

§ 288. **Efficiency of Steam Production.**—The measure of the efficiency of the production of steam is the heat contained in the steam, per unit of weight, compared to the heat contained in the feed-water when it enters the boiler. To make the comparison uniform it is usual to reduce the difference in the amounts of heat in the feed-water and steam, to evaporation from and at 212° Fahr.

Example.—In a boiler 8 lb. steam are generated per pound of coal from feed-water of 110° Fahr., and at a pressure of 140 lb. per square inch absolute. Total heat in the steam at this pressure from Table No. XXII. = 1,189 thermal units per pound. As the temperature of the feed-water is 110° Fahr. $\{1,189-(110-32)\}=1,111$ thermal units have been added to each pound of water in passing through the boiler. From the same table, to generate steam at 212° from water at 212°, 965 thermal units are required; so with 1,108 thermal units $\frac{1,111}{965}=1{\cdot}15$ lb. water from and at 212° can be evaporated, and therefore instead of 8 lb. we should have had 9·20 lb. of steam from and at 212° Fahr. per pound of coal.

The above figures assume that the steam generated is dry satu-

rated steam, and it will therefore only hold where the steam is really dry. Figures giving the results of evaporation (so-called evaporative values, see Table No. 59*) must therefore be received with caution, as serious errors may arise (1) because any water carried over mechanically is credited to the boiler as steam; and (2) because usually it is not possible to determine the temperature of the feed-water in the experiment in question, nor the temperature of evaporation to which the figures were reduced.

§ 289. **Transference of Steam from Boiler to Engine.**—The steam should be taken from the boiler at a position most likely to give dry steam. As evaporation is very rapid in marine boilers, and the level of water is generally much disturbed, due to the motion of the vessel, special steam domes or drums for taking off the steam are often provided; or special steam collectors may be fitted to the boiler, with the object of preventing the water passing over into the steam piping. Steam domes are used in cylindrical and Normand water-tube boilers. In water-tube boilers these arrangements are often insufficient, and an endeavour is made to obtain re-evaporation by throttling down the steam, as for instance in the Belleville boiler. The advantages of throttling the steam are much overestimated.

Example.—Boiler pressure, 200 lb. per square inch absolute; on the engine side of the throttle valve, 100 lb. per square inch absolute; dryness fraction of steam, 0·7, *i.e.*, 70 °/₀ steam, 30 °/₀ water.

Heat contained in the steam per pound prior to passing through the throttle valve (from Table No. XXII.)—

$$353 + 0{\cdot}7\ (844) = 943 \text{ thermal units.}$$

Heat contained in the steam per pound after passing through the throttle valve (from Table No. XXII.)—

$$298 + x_2\ (883) = 943 \text{ thermal units.}$$

As the heat contained in the steam before and after passing through the valve is the same, the dryness fraction of the steam will be—

$$x_2 = \frac{943 - 298}{883} = 0{\cdot}73$$

that is, of the water in the steam, before passing through the valve, only $\frac{{\cdot}73 - {\cdot}7}{3} \times 100 = 10\%$ has been evaporated. It must here be noted that

* In the data given on Table No. 59, the evaporation is given for pressures of from 185 to 230 lb. per square inch above atmosphere, the feed-water not being previously heated.

the expansion of the steam passing through the throttle valve takes place without work being done, and that while steam at a pressure of 200 lb. per square inch is generated, only a pressure of 100 lb. per square inch is available at the engine. By far the best method of drying the steam is to fit separate steam dryers or superheaters into the uptake. On account of the high rate of evaporation demanded in a marine boiler, and the corresponding high temperature of the escaping gases, these dryers partly utilise the heat still available in the escaping gases, and the result is a net gain.

Example.—As above, let the absolute pressure be 200 lb. per square inch, and the dryness fraction of the steam when it leaves the boiler 0·7, evaporation of water per pound coal (from Table No. 59) 8 lb., excess of air 100 °/₀, assuming 13 lb. of air required theoretically per pound of coal, then for every pound of wet steam generated, $\frac{13}{8} \times 2 = 3{\cdot}25$ lb. of air are required. Temperature of the escaping gases before entering the superheater 662° Fahr., after leaving the heater 392° Fahr. Then per pound of wet steam $270 \times 0{\cdot}22 \times 3{\cdot}25 = 190$ thermal units will be available. With these units we can evaporate $\frac{190}{845} = 0{\cdot}23$ lb. of steam at a pressure of 200 lb. per square inch absolute. As 0·3 lb. of water is present per pound of wet steam, there would only be $0{\cdot}3 - 0{\cdot}23 = 0{\cdot}07$ lb. of water in the steam leaving the heater, that is $x = 0{\cdot}93$.

The construction and heating surface of the superheater is chiefly determined in marine boilers by practical considerations, among which the chief are space available and ease of access. In general it may be theoretically of advantage to select a type having as many small tubes as possible, with very thin walls, because the conditions would then be the most favourable for the rapid transmission of heat. In any case it is of course essential to cover the exteriors of the steam pipes with non-conducting materials, to diminish losses by radiation. It must, however, be admitted that up to the present it has not been found practicable to use superheaters and steam dryers in connection with marine engines, the extra complication to the piping not being counterbalanced by any corresponding marked advantage.

§ 290. **Percentage of Water in Steam.**—With the great demands made on modern marine engines, and having regard to the great extent to which the boilers must under certain conditions be forced, it is advisable to test the steam for its *percentage of water* before it enters the engine. The methods hitherto in use are not very practical, with the single exception of Peabody's throttling calorimeter, which is adapted for use with marine boilers (see Part VI.).

SECTION II.

CYLINDRICAL BOILERS.

§ 291. **General Remarks.**—Direct tube type cylindrical boilers, or Admiralty boilers, are only used for small engines, or where the available head room is much restricted (Figs. 421, 422). Otherwise the return tube type is always employed. (See Figs. 428 to 435.) They are made either single-ended (Fig. 434) or double-ended (Figs. 428, 429). Single-ended boilers have a heating surface up to 3,000 square feet, double-ended up to 6,000 square feet. According to the size of the boiler, single-ended have from one to four furnaces, double-ended from four to eight furnaces. The height of the steam space (from the crown of the boiler to the lowest water level) is about 0·225 to 0·25 of its internal diameter. The higher it is, the larger will be the water surface, other conditions being the same, and the greater the capacity of the boiler for furnishing dry steam. With comparatively small boilers, horizontal or vertical steam drums connected to the boiler shell are often provided, so that the steam may be taken off as far away as possible from the surface of the water. The drum is generally connected to the boiler through an opening of about 16 inches diameter.

§ 292. **Selection of Heating Surface and Grate Area.**—The amount of heating surface is determined by the type of engine selected, and the quantity of steam required to drive it, the method of firing the boiler (whether with or without forced draught), and the class of fuel used. To secure economy of coal consumption natural draught is best, and it is therefore most generally used in merchant ships. But where the consumption of coal at maximum power is of less importance, and the main considerations are economy of space and weight, as in warships, forced draught of a greater or less degree is almost always employed.

Table No. 60, page 476, gives the usual power developed per square foot of grate surface, in cylindrical boilers with natural draught, while Table No. 61 gives these ratios for various types of marine cylindrical boilers. In calculating the dimensions of the boiler, the grate area, Table No. 60, is always taken as the basis, and from the ratio H : R, Table No. 61, the heating surface is determined.

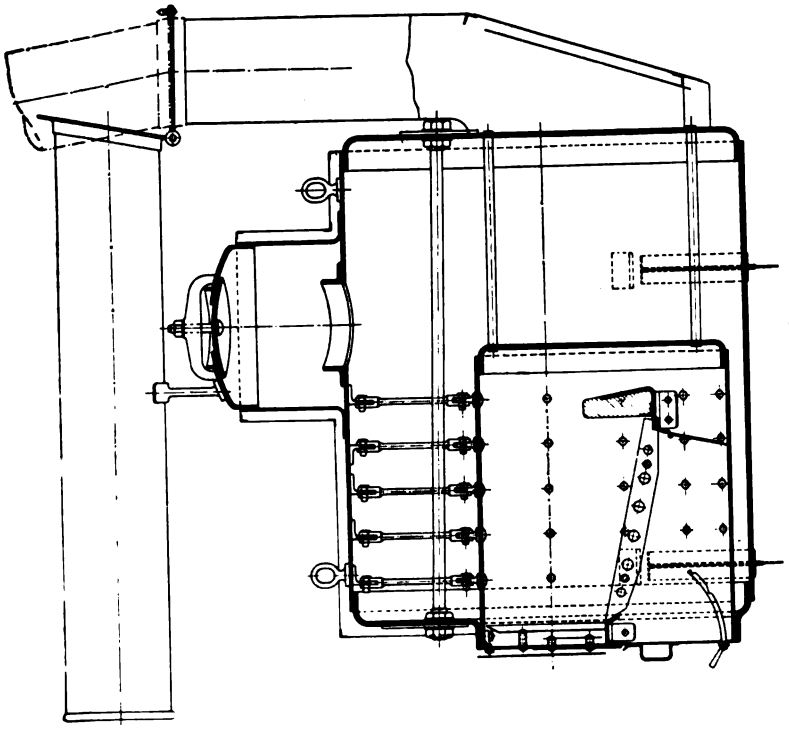


Fig. 422.

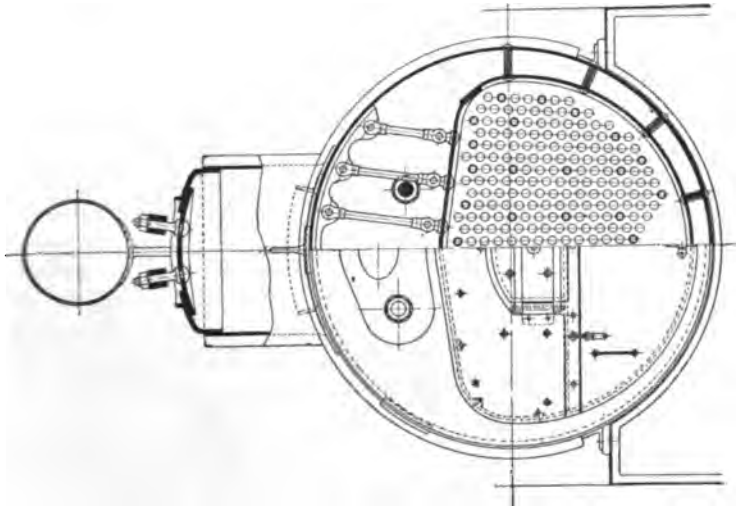


Fig. 421.

TABLE No. 60.

*Maximum Horse-power developed per square foot of Grate Surface with Natural Draught.**

		Type of Engine used.								
		Compound.			Triple Expansion.			Quadruple Expansion.		
Steam pressures—pounds per square inch - - -		130	145	160	175	185	200	215	230	245
Height of Funnel above Grate	Feet.									
,,	26–33	7·4	7·7	8·0	8·4	8·5	8·7	9·0	—	—
,,	33–50	7·8	8·0	8·4	8·7	9·1	9·3	9·4	—	—
,,	50–65	8·0	8·3	8·7	9·1	9·4	9·7	10·0	10·4	10·7
,,	65–82	8·4	8·8	9·2	9·7	10·1	10·5	10·8	11·2	11·5
,,	82–98	—	—	—	10·2	10·6	10·9	11·3	11·8	12·2
,,	98–115	—	—	—	10·7	11·1	11·6	12·0	12·5	13·0

TABLE No. 61.

Heating Surface and Area of Grate in Cylindrical Boilers.

Type of Ship.	Kind of Draught used.	Type of Engine.	Consumption of Coal † per I.H.P. per hour, pounds.	Heating Surface per I.H.P. in square feet.	Ratio $\frac{H}{R} = \frac{H.S.}{G.S.}$
Merchant steamers	Natural	Compound	2·2 —2·6	3·6—4·4	26—35
Do.	Do.	Triple	1·62—1·76	3·2—3·6	30—35
Do.	Do.	Quadruple	1·43—1·65	3·2—3·6	32—35
Do.	Slightly forced	Do.	1·65—1·87	2·2—3·2	36—40
Do.	Howden	Do.	1·65—1·87	2·2—3·1	36—40
Ironclads and heavy cruisers	Moderately forced	Do.	2·0 —2·31	1·8—2·7	25—30
Light cruisers	Forced	Do.	—	1·6—2·2	28—32

§ 293. **Furnaces and Grates.**—Furnaces may be made smooth, ribbed, or corrugated. If they are plain and not corrugated, their thickness must be relatively large (especially in the case of large furnaces for high pressures); hence they are generally only used for small boilers, and if their diameter exceeds 2 feet 8 inches, they are strengthened by what are known as "Adamson" rings, Fig. 422A. For the high pressures used in triple and quadruple expansion engines, corrugated furnaces are almost exclusively employed.

* These powers may, with skilful stoking, be maintained for six hours, and under favourable conditions of wind, &c., for several days.

† The smaller rates of coal consumption are for larger boilers.

For Morrison corrugated furnaces see Fig. 423.
„ Fox's „ „ „ 424.
„ Purves' „ „ „ 425.
„ Deighton's „ „ „ 426.

The back of the furnace should be connected to the tube plate by one or other of the methods shown in Figs. 427, 428, or Figs. 430, 432, where allowance is made for the unequal expansion of the furnace. Sometimes these joints, or even the whole of the lower part of the fire box above the bridge, are fitted with fire-brick masonry, as shown at Fig. 430.

Fig. 422A.

It is advisable to fit the furnace in in such a way that it can be taken out when necessary, and a new one put in. For this purpose the opening in the front end plate must be rather larger than the external diameter of the furnace, Fig. 427, and the back flange must be so shaped that it can be drawn through this opening. Sometimes the back end of the corrugated furnace is drawn in and flanged, in order to facilitate its removal.

The length of the grate does not as a rule exceed $6\frac{1}{2}$ feet (78 inches), because a greater length presents difficulties in the way of stoking.

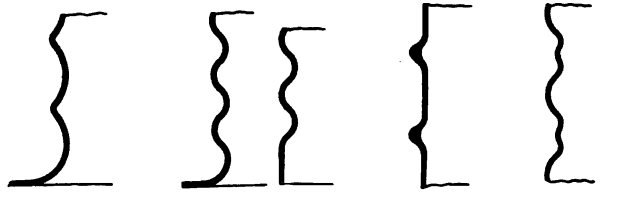


Fig. 423. Fig. 424. Fig. 425. Fig. 426.

The length is about twice to two and a half times the breadth (internal diameter) of the furnace. Where possible the latter should not be less than 27 inches, and in very large boilers it should not exceed 50 inches.

Number of furnaces in single-ended boilers—

For diameters up to about 9 feet = 1 furnace
„ „ „ 13 „ = 2 „
„ „ „ 15 „ = 3 „
„ „ above 15 „ = 4 „

Double-ended boilers should have twice as many furnaces.

The grate consists of one or of two sets of bars according to its length, the ends of which rest on the dead-plate and on the bearers of

the fire bridge respectively, and when two sets of bars are fitted the centre is supported on bearers. (See Figs. 427 to 432.)

Length of the cast-iron or wrought-iron fire bars from 20 to 50 inches, sometimes as much as 5 feet 6 inches.

Width of the cast-iron or wrought-iron fire bars at the top about 0·5 to 1 inch.

Space between the bars about $\frac{3}{8}$ to $\frac{1}{2}$ inch.

The fire bars must have sufficient "play," lengthways, as well as sideways, to allow them to expand freely, as they get hot. With corrugated

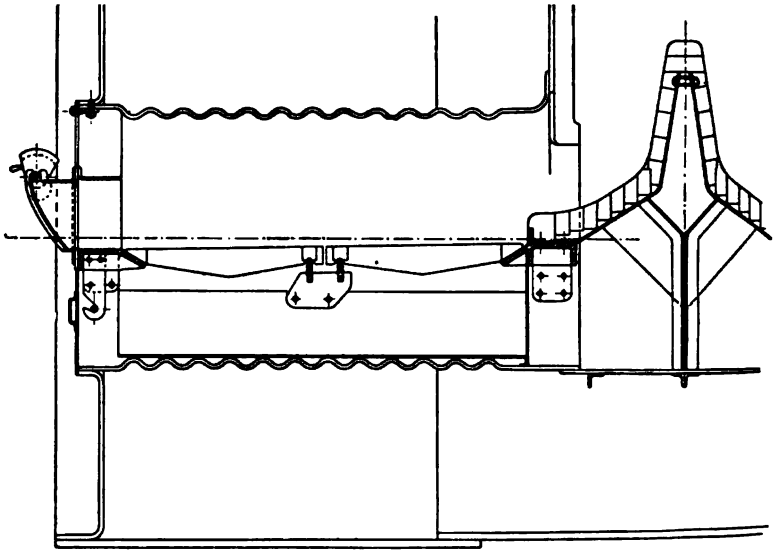


Fig. 427.

furnaces special side bars are fixed at the side of the grate, which should fit accurately into the corrugations of the furnace. The surface of the grate is generally somewhat inclined towards the fire bridge, so that, if the latter is fairly high (about 5 to 8 inches), there may still be a sufficiently large space left above it.

With very large boilers having three or four furnaces, the grates in the two side or wing furnaces are, for convenience of stoking, sometimes placed rather lower than the grate of the centre furnace. (See Fig. 427.) The furnace fronts and doors are generally made of sheet iron, or of cast iron if induced draught or Howden's forced draught is used, and fitted on the inside with baffle plates

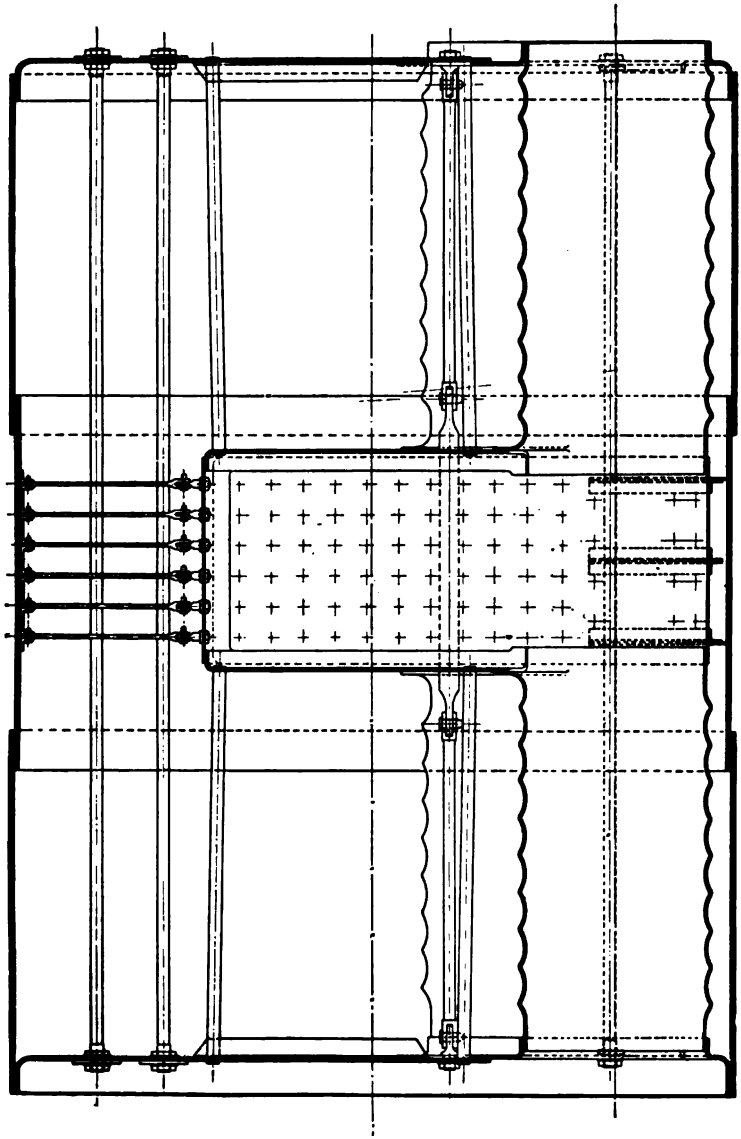


Fig. 428.

to prevent them from getting burnt. The usual size of the opening of fire doors is from 16 by 12 inches up to 18 by 14 inches. Instead of fire doors which open sideways, those opening upwards and inwards (see Figs. 427 and 430) or outwards are often used. They are provided with counterweights to make them open and keep open with ease.

The fire bridge is made about 5 to 8 inches above the level of the grate at the back end, and is either horizontal on the top, or rises a little,

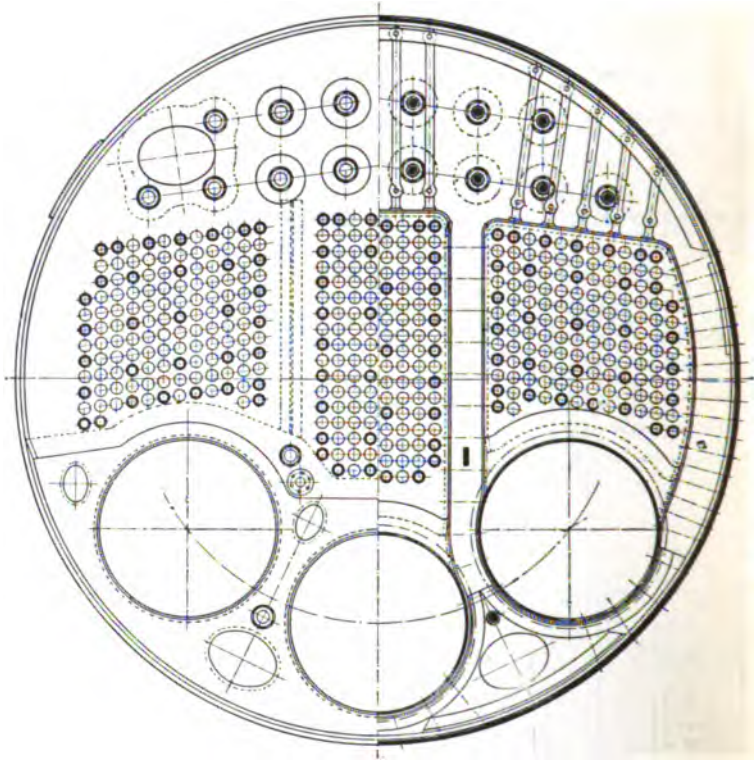


Fig. 429.

to guide the gases of combustion upwards. The clear height of the flue above the fire bridge is about $0{\cdot}4d$ to $0{\cdot}45d$, d being the internal diameter of the furnace.

§ 294. **Boiler Tubes.**—The external diameter of these varies from 2 to $3\frac{1}{2}$ inches, their thickness from ·1 to ·15 inch. Thickness of stay tubes from ·2 to ·4 inch, according to size of tube.

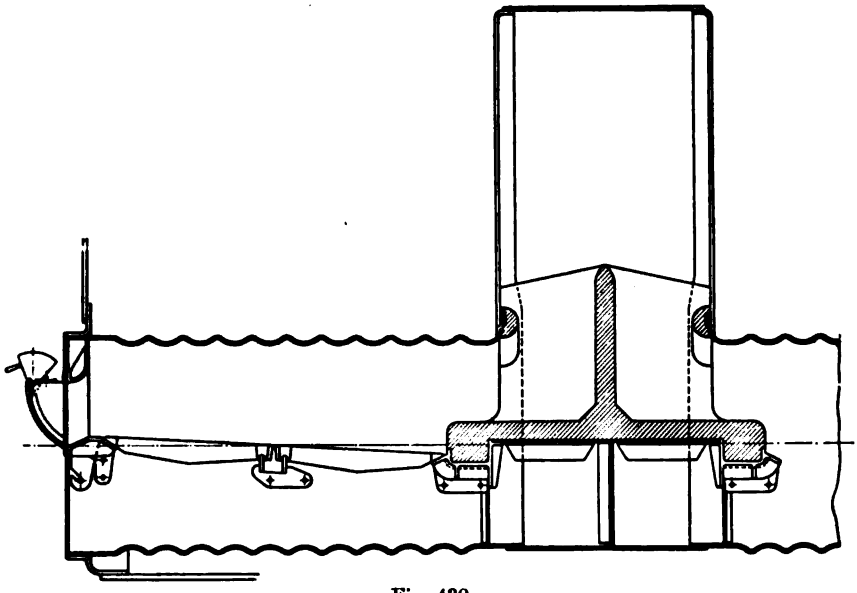


Fig. 430.

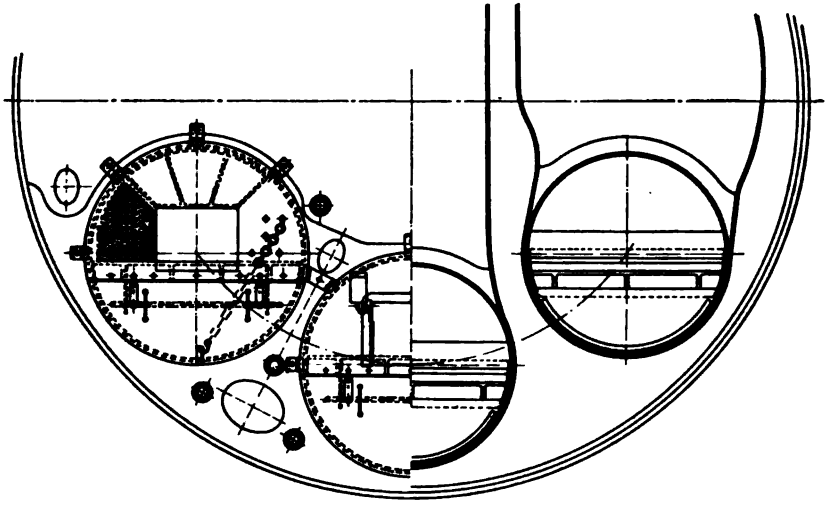


Fig. 431.

Pitch of the tubes :—

With an external diameter of 2 inches, the pitch is $2\frac{5}{8}$ to $2\frac{7}{8}$ inches.

,,	,,	$2\frac{1}{4}$	,,	,,	3	,,	$3\frac{1}{8}$	,,
,,	,,	$2\frac{1}{2}$	,,	,,	$3\frac{1}{4}$	,,	$3\frac{3}{8}$	,,
,,	,,	$2\frac{3}{4}$	,,	,,	$3\frac{5}{8}$	,,	$3\frac{3}{4}$	,,
,,	,,	3	,,	,,	$3\frac{7}{8}$	,,	4	,,
,,	,,	$3\frac{1}{4}$	,,	,,	$4\frac{1}{8}$	,,	$4\frac{1}{4}$	,,
,,	,,	$3\frac{1}{2}$	,,	,,	$4\frac{3}{8}$	,,	$4\frac{1}{2}$	,,

The length of boiler tubes, with natural draught, is equal to about twenty-three to thirty times the external diameter of the tube; and with

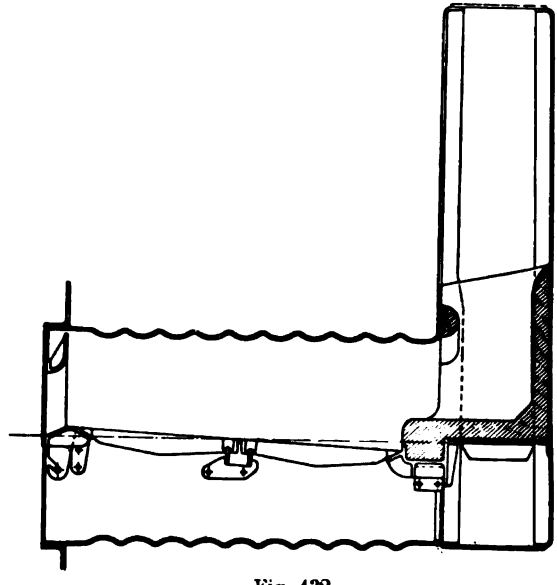


Fig. 432.

artificial draught, about thirty-five to forty times the external diameter of the tube. Boiler tubes are generally merely expanded into the smooth-drilled holes in the tube plates, while the stay tubes for stiffening them are first screwed with a fine thread into both ends, and then expanded. Sometimes the boiler tubes, as well as the stay tubes, are turned over and beaded. The holes for the boiler and stay tubes in the front tube plates are made about $\frac{1}{8}$ inch larger than the external diameter of the tube, so that the tubes may be easily drawn out from the front. The front ends of the tubes are correspondingly increased in size.

§ 295. **Manholes** are made in the shell, in the steam drum, or in the front end of the boiler, and should not be less than about 12 by 16 inches. In exceptional cases they may be 11 by 15 inches. Inspection holes in the front end and mud holes in the lower part of the boiler are made smaller. All openings for manholes and mud holes must be stiffened, either by stiffening rings riveted on, or by flanging the edges of the openings inwards. The smooth inner edge of these flanges is generally made to serve as the jointing surface of the corresponding

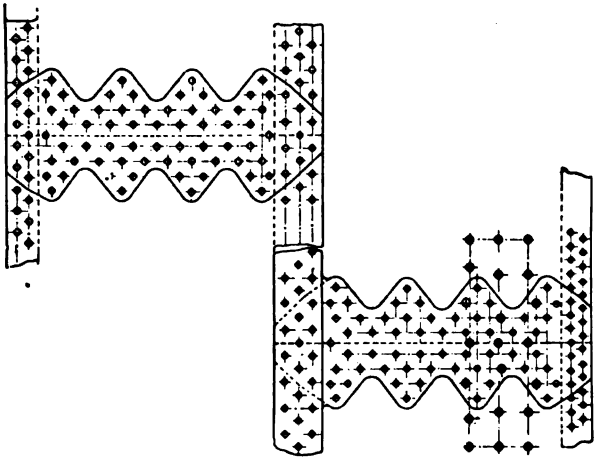


Fig. 433.

cover. If the manhole is in the shell, its smaller axis must run longitudinally to the boiler, and the opening must be strengthened by a ring riveted on, either inside or out, to make good the material cut away.

§ 296. **Thickness of Material Used.**—For marine boilers this is generally calculated according to the regulations laid down by the Insurance Companies or Classification Societies. The principal rules laid down by some of these Companies, and the Hamburg Rules of 1898, are as follows.

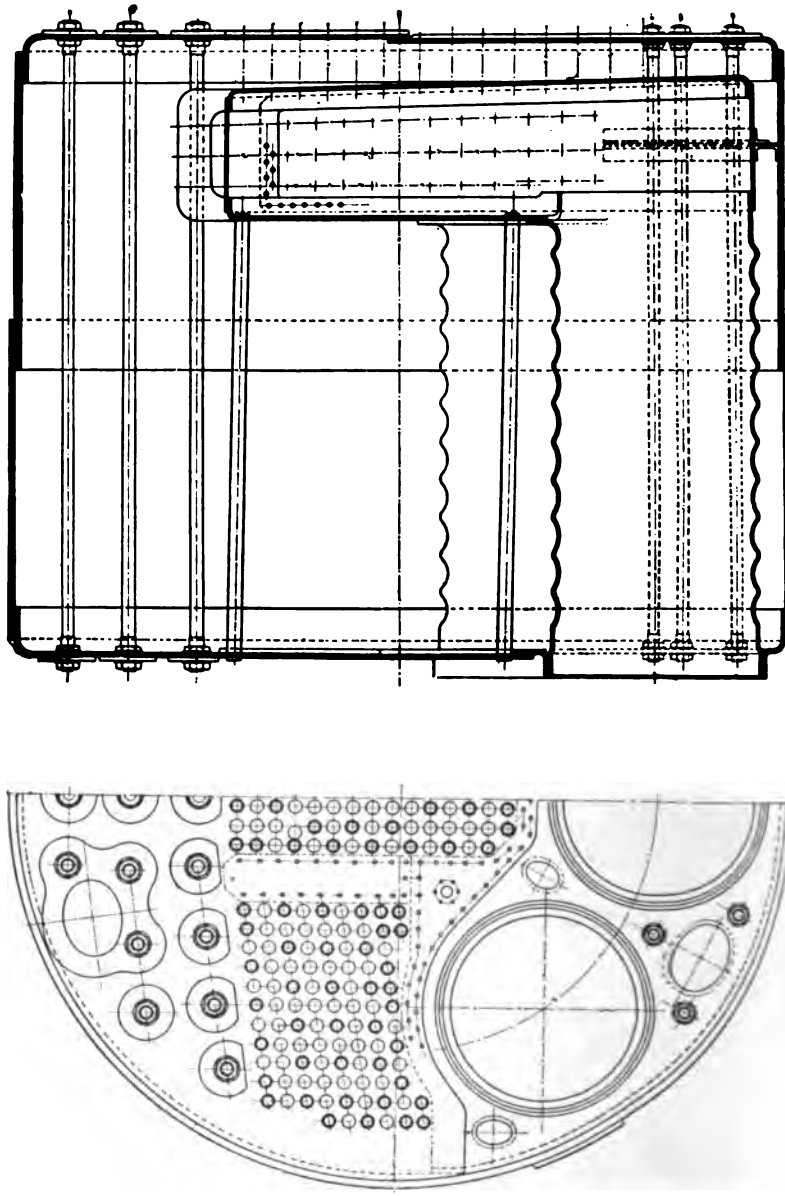


Fig. 434.

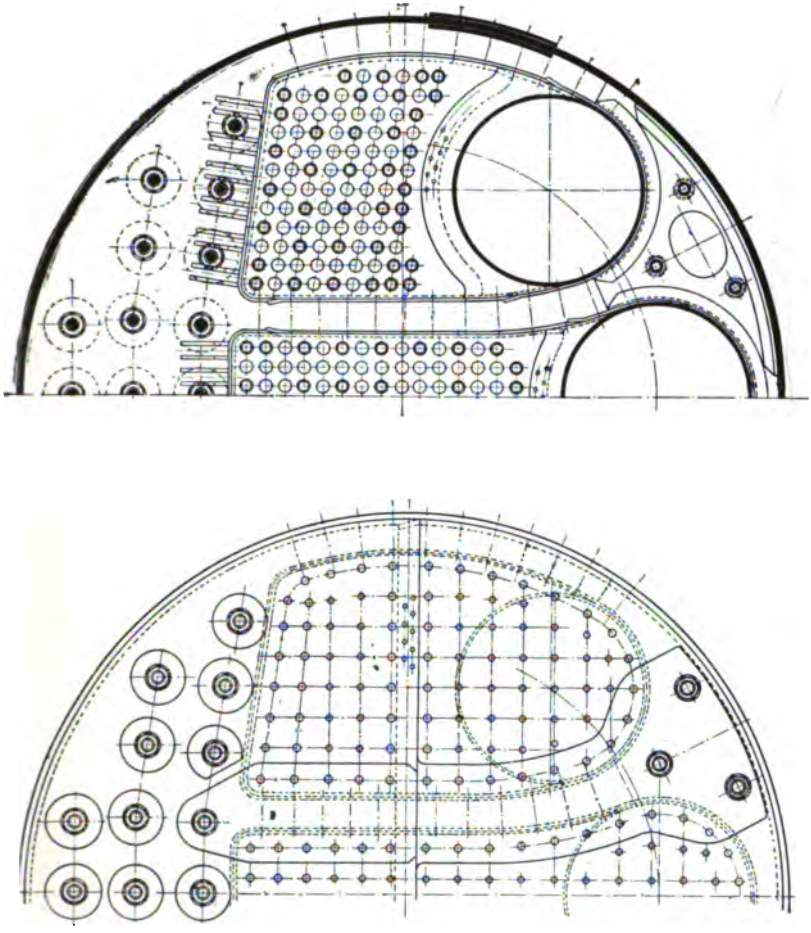


Fig. 435.

Fig. 436.

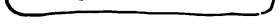


Fig. 437.

Fig. 438.

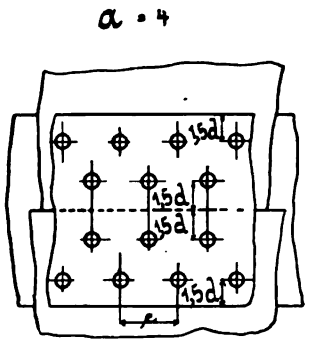


Fig. 439.

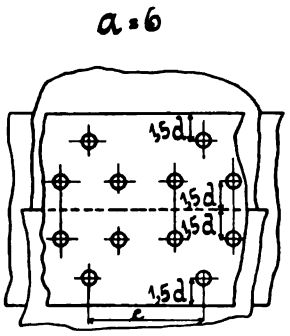
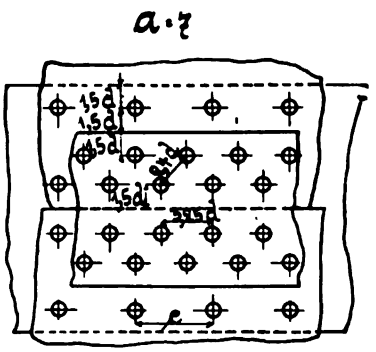


Fig. 440.



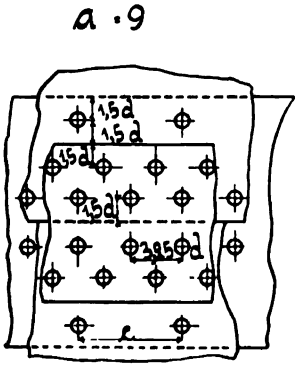


Fig. 441.

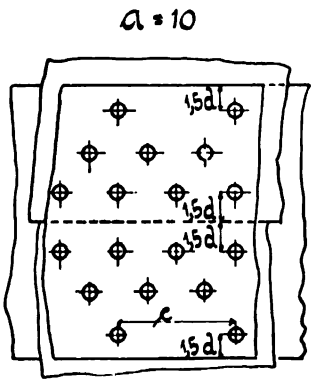


Fig. 442.

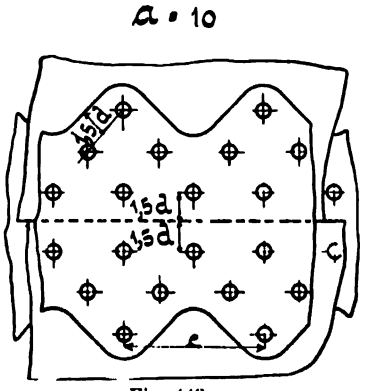


Fig. 443.

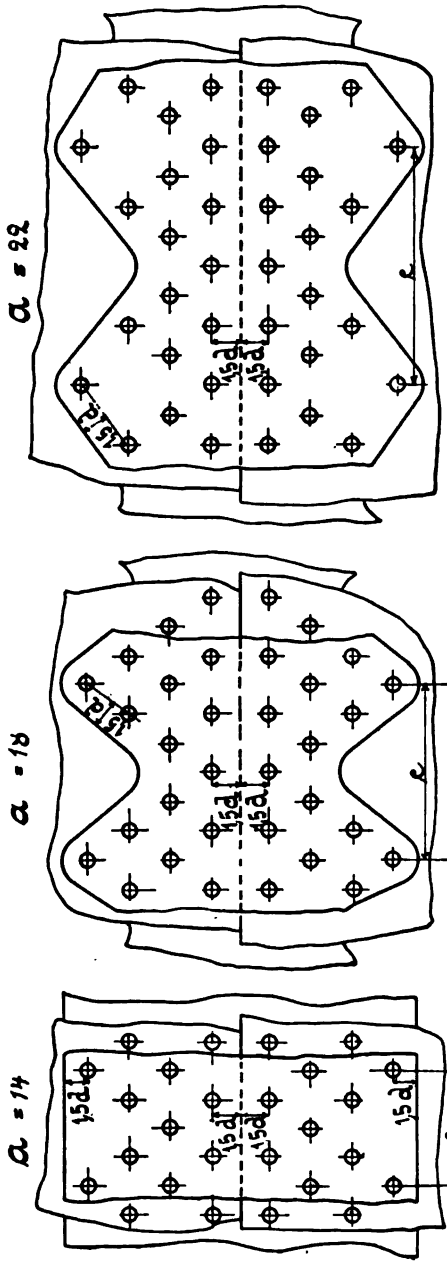


Fig. 444.

Fig. 445.

Fig. 446.

§ 297. German Lloyd's Rules.

(The metric system has been retained for these rules in order to prevent complication of the formulæ.)

(a.) Boiler Shells and Steam Domes tested for Internal Pressure.— The thickness of the plates and the sectional area of the rivets in a longitudinal joint are calculated according to the following formulæ:—

$$1. \qquad S = \frac{P \cdot D}{2} \times \frac{b}{B} \times \frac{e}{e - d}$$

$$2. \qquad \frac{d^2 \pi}{4} = \frac{P \cdot D}{2} \times \frac{n}{N} \times \frac{e}{a}$$

Here S = thickness of the plate in centimetres.

P = working pressure above atmosphere in kilogrammes per square centimetre.

D = maximum internal diameter in centimetres.

B = tensile strength of the plate in kilogrammes per square centimetre.

N = shearing stress of the rivets in kilogrammes per square centimetre.

b = working factor of safety for the boiler plate.

n = working factor of safety for the rivets.

e = pitch of the rivets in centimetres.

a = sectional area of all the rivets in a row.

d = diameter of the rivets in centimetres (compare Figs. 436 to 446).

The values allowed for B and N by the constructor must be the minimum values when tested for strength (see Table XXXIV., Part VIII.), and must be given on the drawing submitted to the German Lloyd's for approval.

The shearing stress N of the material of the rivets, if other values have not been determined by direct tests, is taken at 0·875 of the tensile strength in wrought-iron rivets, and 0·8 of the tensile strength in mild steel rivets.

The factor of safety b of the plate is to be taken at 5, but may be reduced to 4·75 if the rivet holes in the longitudinal seams are drilled and the riveting done by machinery, and to 4·5 if, in addition to this, the longitudinal seams are in double shear. If the plate is more than ½ inch thick, the circumferential seams must be double riveted; if it is 1 inch thick or more, the middle circumferential seams must be treble riveted. The diameter of the rivet must be at least equal to the mean thickness of the two plates it connects. If stay bolts are attached to the

shell plates the strength of the rows of stay bolts must not be less than that of the longitudinal seams of the body.

The factor of safety n for the material of the rivets is to be taken as equal to b with lap joints; if the rivets have double butt straps, as equal to $1{\cdot}15b$. In the latter case it may be taken as equal to b if the rivet holes are drilled after the plates are fitted together and the joints riveted by machinery. The thickness of the butt straps must be at least 0·75 that of the plate. In boilers having no middle circular joint, the outside butt straps must not be thinner than the shell plate, unless the riveting is done by hydraulic pressure. With thin boiler plates the diameter of the rivets d must not exceed $2s$, with thick plates it must not be taken at less than s. With lap joints and straight-sided butt straps the pitch of the rivets e must not be more than eight times the thickness of the plate or of the strap. If the butt straps are straight, the pitch of the rivets must not exceed $e = \mathrm{c} \cdot s + 4{\cdot}1$ centimetres, c being a constant, taken from the following table:—

Table of Values of c.

Number of Rivets in one Row.	c for Lap-jointed Longitudinal Seams.	c for Longitudinal Seams in Double Shear.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

If the pitch is more than 254 millimetres, the butt-strap riveting must be zigzag. (See Figs. 433, 443, 445, and 446.) With double-riveted joints the pitch of the rivets must not exceed $3{\cdot}75d$; with zigzag-riveted joints the diagonal distance between two rivets must not be less than $2{\cdot}4d$. The strength of welded seams in wrought-iron boilers is taken at 0·7 of that of the solid plate. Stiffening rings round manholes, &c., must have a sectional area sufficient to counterbalance the weakening produced in the plate.

(*b*.) *Flat Plates.*—For flat boiler plates stiffened with gusset stays or stay bolts, the thickness of the plate is determined from the formula—

$$s = \mathrm{c} \cdot l\sqrt{\mathrm{P}} \text{ for steel plates.}$$
$$s = 1{\cdot}12\ \mathrm{c} \cdot l\sqrt{\mathrm{P}} \text{ for wrought-iron plates.}$$

Where s = thickness of the plate in centimetres.
l = distance between the stays in centimetres.
P = pressure above atmosphere in kilogrammes per square centimetre.
C = constant, with the following values :—

C = 0·024 if the plates are in contact with the hot gases and the water, and the stays are screwed in, and riveted over.
C = 0·022 if the plates are in contact with the hot gases and the water, and the stays are screwed in, and fitted with nuts outside.
C = 0·021 if the plates are not in contact with the hot gases, and the gusset stays and stay bolts are screwed in, and have riveted heads.
C = 0·020 if the plates are not in contact with the hot gases, and the gusset stays and stay bolts are screwed in, and fitted with nuts.
C = 0·018 if the plates are not in contact with the hot gases, and the diameter of the washer is $0{\cdot}4l$, and its thickness $0{\cdot}667s$.
C = 0·017 if the plates are not in contact with the hot gases, and the diameter of the washer is $0{\cdot}6l$, and its thickness $0{\cdot}833s$.
C = 0·016 if the plates are not in contact with the hot gases, and the diameter of the washer is $0{\cdot}8l$, and its thickness equal to s.

The thickness of the flat plates in the vicinity of the nest of boiler tubes is determined from the same formula, l being the mean distance between the stay tubes, taken from centre to centre, and C = 0·020.

To determine the thickness of the flat plates between the nests of the boiler tubes, we must take

l = distance from centre to centre of the bounding rows of tubes in centimetres.
C = 0·022 if in the bounding row of tubes every third tube is a stay tube.
C = 0·020 „ „ „ every second tube „
C = 0·019 „ „ „ each tube is a stay tube.

If the top of the combustion chamber is not connected by ties, &c., to the boiler shell, but supported on girders which project beyond the edges of the tube plates, the thickness of the latter must not be less than

$$s = \frac{P \times W \times b}{1800(b - d)}$$

Where s = thickness of tube plate in cm.
W = width of combustion chamber in cm.
b = distance between the boiler tubes from centre to centre.
d = inside diameter of the boiler tubes in cm.
P = allowable boiler pressure in kilogrammes per sq. cm.

Boiler plates which are in contact on one side with the hot gases, and on the other with the steam, must be 10 per cent. thicker than the calculated value, and it is also advisable to protect them by baffle plates.

(*c.*) *Iron or Steel Furnaces.*—The thickness of furnace plates is determined from the following formula—

$$s = 0{\cdot}00385\sqrt{\text{P . D . L}}$$

Where s = thickness of plate in centimetres.

P = working pressure above atmosphere in kilogrammes per square centimetre.

D = external diameter of the furnace in centimetres.

L = length of the furnace in centimetres, or if stiffening rings are used, greatest distance between two rings.

The thickness of the plate s must not be less than that given by the following formulæ. With ribbed furnaces (Purves' patent), D denotes the external diameter of the flat part between the ribs; and with corrugated furnaces, it denotes the external diameter at the lowest part of the corrugation.

$s = \frac{\text{P . D}}{740} + 0{\cdot}3$: for plain furnaces without Adamson rings.

$s = \frac{\text{P . D}}{900} + 0{\cdot}3$: for plain furnaces with one Adamson ring. The distance between the ring and the plate must not exceed 1·22 metres.

$s = \frac{\text{P . D}}{1010} + 0{\cdot}3$: for plain furnaces with two Adamson rings. Distance between the rings not to exceed ·79 metre.

$s = \frac{\text{P . D}}{1125} + 0{\cdot}3$: for plain furnaces with three Adamson rings. Distance between the stiffeners not to exceed ·61 metre.

$s = \frac{\text{P . D}}{1220} + 0{\cdot}3$: for corrugated furnaces (Fox, Morrison, or Deighton's patent) having a tensile strength of 35 to 41 kilogrammes per square millimetre (22 to 26 tons per square inch). The thickness of corrugated furnaces should be at least 0·8 centimetre (say $\frac{3}{8}$ inch), height of corrugation at least 3·8 centimetres ($1\frac{1}{2}$ inch), and length of the plain end not more than 25·4 centimetres (10 inches).

$s = \frac{\text{P . D}}{1220} + 0{\cdot}3$: for plain furnaces with four Adamson rings, in which the space between the rings does not exceed 1 foot 8 inches, and also for ribbed furnaces (Purves' patent). The height of the ribs must not be less than 3·4 centi-

metres ($1\frac{3}{8}$ inch) above the plain part, depth of the inner grooves not more than 1·9 centimetre ($\frac{3}{4}$ inch), length between the ribs not more than 22·9 centimetres (9 inches), and length of plain end not more than 15·2 centimetres (6 inches).

(*d.*) *Stays and Stay Bolts or Screw Stays.*—The stress upon welded iron stays must not be more than 0·1, and with iron or steel non-welded stays not more than 0·143 of the tensile strength of the material, but the following limits must not be exceeded :—

For wrought-iron welded stays, 350 kilogrammes per square centimetre (5,000 lb. per square inch).
For wrought-iron non-welded stays, 500 kilogrammes per square centimetre (7,000 lb. per square inch).
For mild steel, 600 kilogrammes per square centimetre (8,500 lb. per square inch).

(*e.*) *Girders for supporting Tops of Combustion Chambers.*—The girders for the flat tops of combustion chambers are calculated from the following formulæ—

$$b = \frac{P(W - l)e \cdot L}{C \cdot h^2} \text{ for wrought iron.} \qquad b = 0{\cdot}9\,\frac{P(W - l)e \cdot L}{C \cdot h^2} \text{ for steel.}$$

Here W = width of combustion chamber in centimetres.
P = working pressure above atmosphere in kilogrammes per square centimetre.
l = distance of the stays from each other in the girder, or if there is only one stay, half the length of the girder, in centimetres.
e = spacing between the girders in centimetres.
L = length of the girders in centimetres.
h = height „ „
b = thickness „ „
C = 420, if there is one stay in each girder.
C = 630, if there are two or three stays in each girder.
C = 720, if there are four stays in each girder.

The ends of the girders must fit on to the vertical end walls of the combustion chamber, and must project about 4 centimetres ($1\frac{1}{2}$ inch) above the top.

(*f.*) *Donkey Boilers.*—As far as the rules already laid down for the construction of boilers are applicable, they apply also to donkey boilers.

(*g.*) *Manufacture of the Boiler.*—This can only be efficiently carried

out, and the coefficient of safety for the strength of the shell plates, as already mentioned, be determined, if the following conditions are complied with:—

The preparing and working of the material, such as bending, dishing, and flanging the plates, drilling the holes, &c., must be done with the greatest care, and in a satisfactory manner. If the rivet holes do not exactly coincide, they must be rimered with the plates in place. Riveting the joints and caulking the seams must be done as carefully as possible. If the holes are punched, they must be enlarged sufficiently to make good the damaged metal round them. If the edges of the plate are torn, or the rivets defective, they must be rejected, and sound plates and rivets substituted for them. All seams must, if possible, be caulked inside and out.

The shell plates of cylindrical boilers must be bent with the grain. The butt straps must be cut from plates of the same quality as the shell plates, and the grain should run in the same direction as that of the plates. If single butt straps are used, they must be $\frac{1}{8}$ inch thicker than the boiler plates.

All openings for manholes, steam domes, &c., must be strengthened with riveted wrought iron or steel rings of flat, angle, or T section, or better by dishing the plates, so that the weakening of the plate produced by the hole is completely made good. If there are mud holes in the bottom end plates, and these are for other reasons made stronger than the rest of the boiler, it is not necessary to compensate for the weakening of the plate. All the larger fittings must be secured by means of studs or set screws to thick faced seatings or flanges riveted to the sides of the boiler. These studs must not penetrate the boiler plate. Steel stays should not be welded. If rivet holes are drilled in a boiler with the plates roughly erected together, it is advisable, after the holes have been made, to take the plates apart, and remove the burr. Angle joints should as far as possible be made by dishing the plates. To rivet by hydraulic pressure is generally better than to rivet by hand.

It depends chiefly on the material used, and its treatment, whether the steel plates should be annealed, and this is a question which must be left to the judgment of the boilermaker; but intense heating of the plate locally should in any case be avoided. The material to be worked up must comply with the conditions laid down in the rules for testing wrought iron and steel to be used in boilers. (Compare Table XXXIV., Part VIII.)

§ 298. **Hamburg Standard, 1898.**—Extract from the rules for

calculating the strength of material for new steam boilers (here also the metric system has been retained):—

1. Boiler Shells.

The thickness of the plate of the boiler shell is $s = d\frac{p \cdot x}{200 K_2}$

Here s = thickness of plate in millimetres.

d = inside diameter in millimetres.

p = maximum working pressure above atmosphere, kilogrammes per square centimetre.

x = factor of safety.

K = tensile strength of the material, kilogrammes per square millimetre.

z = strength of riveted joint, as compared with strength of whole plate.

The thickness of the plate must not be less than 7 millimetres.

The factor of safety x of the plate must not be less than 4·5 at the weakest part.

If the joints are in double shear, the factor of safety may be reduced to 4, but this assumes the boiler to have been most carefully constructed in every way. It should be considered whether, to conform to local working conditions, additional thickness is required, and this is necessary if the calculation gives a thickness of plate of less than 10 millimetres.

The strength of material to be used is determined by the Würzburg Standard of 1895.

I. *Wrought Iron* (*Tensile Strength*).

	Furnace Plates.		Dished End Plates.		Shell Plates.	
	With the Grain.	Across the Grain.	With the Grain.	Across the Grain.	With the Grain.	Across the Grain.
Number denoting quality of metal or quality-factor -	56	49	50	45	43	38
Tensile strength in kilogrammes per sq. mm. - -	36	34	35	33	33	30
tons per sq. in. -	22·8	21·5	22·2	21	21	19
Elongation per cent. on 200 mm. (say 8 inches) - -	20	15	15	12	10	8

II. *Mild Steel* (*Tensile Strength*).

	Furnace Plates.		Dished End Plates.		Shell Plates.	
	With the Grain.	Across the Grain.	With the Grain.	Across the Grain.	Across the Grain.	With the Grain.
Number denoting quality of metal or quality-factor -	62	62	61	61	60	60
Tensile strength in kilogrammes per sq. mm. - -	34–40	34–40	36–42	36–42	39–45	39–45
tons per sq. in. -	21·5–25·3	21·5–25·3	22·8–26·6	22·8–26·6	24·7–28·5	24·7–28·5
Elongation per cent. on 200 mm. (say 8 inches) - -	25	25	22	22	20	20

If the thickness of the plates is calculated on the basis of a greater tensile strength than the minimum values given in the Würzburg Standard, proof may be required that the plates really are of the strength which forms the basis of calculation.

The seams must be riveted in such a way that they will withstand all tendency to slip, and the resistance of the rivets to shearing stress must not be less than the strength of the plates allowed for in the riveted joint. The maximum stress in a rivet in kilogrammes per square millimetre and in tons per square inch must not exceed—

For a single riveted lap joint { 7·0 kilogrammes per square millimetre. / 4·44 tons per square inch.

„ double „ „ { 6·5 kilogrammes per square millimetre. / 4·12 tons per square inch.

„ treble „ „ { 6·0 kilogrammes per square millimetre. / 3·8 tons per square inch.

„ single riveted double butt strap joint { 12·0 kg. per sq. mm. / 7·6 tons per sq. in.

„ double „ „ „ { 11·5 kg. per sq. mm. / 7·3 tons per sq. in.

„ treble „ „ „ { 11·0 kg. per sq. mm. / 7·0 tons per sq. in.

If butt straps are used for the joints, they must be cut from plates of at least the same quality as the shell plates.

2. FURNACES.

The thickness of these should be—

(*a.*) With plain furnace plates—

$$s = \frac{p \cdot d}{2{,}000}\left(1 + \sqrt{1 + \frac{a}{p} \times \frac{l}{l + d}}\right) + c.$$

Here s = thickness of plate in millimetres.

p = maximum working pressure above atmosphere in kilogrammes per square centimetre.

d = internal diameter of furnace in millimetres.

l = length of furnace, namely, greatest distance of the stiffeners from each other, in millimetres.

$a = 100$ for horizontal furnaces with lap-jointed longitudinal seams.

$a = 80$ for horizontal furnaces with butt straps or welded longitudinal seams.

$c = 1{\cdot}5$ millimetre if p is as much as 70 lb. per square inch.

$c = 1{\cdot}0$ " " " 85 " "

$c = 0{\cdot}5$ " " " 100 " "

$c =$ zero if p is above 100 " "

(*b.*) With corrugated and ribbed furnaces, if in the latter the distance of the ribs is 9 inches apart, then—

$$s = \frac{p \cdot d}{1{,}000} + c$$

taking c at 3 millimetres.

The thickness of plate must not be taken at less than 7 millimetres.

3. Flat Surfaces.

(*a.*) *Flat Plates.*—The thickness of these should be—

$$s = 1{\cdot}5 + 0{\cdot}1e\sqrt{\frac{p \cdot c}{\kappa}}$$

Where s = thickness of plate in millimetres.

p = maximum working pressure above atmosphere in atmospheres.

e = distance of the stays or screw stays from each other in millimetres.

κ = tensile strength of the material in kilogrammes per square millimetre.

$c = 1{\cdot}323$, if the stays or screw stays are screwed and riveted into the plates.

$c = 1{\cdot}0314$, if they are screwed into the plates and fitted with a nut on the outside.

$c = 0{\cdot}9774$, if they are screwed into the plates, and fitted inside and out with nuts and washers, the diameter of which must be at least equal to four-tenths the distance between the stays or rows of stays. Thickness of washers at least $\frac{3}{5}s$, and it must be increased if the diameter of the

washers is more than 1·5 times the diameter of the nuts, measured across the corners.

$c = 0{\cdot}8658$, if the stays or screw stays on each side of the plate are fitted with nuts and washers, and the outside washer is riveted to the plate, and has a thickness of at least $0{\cdot}75s$ and a diameter of at least $0{\cdot}6e$.

(*b.*) *Dog Stays on Flat-bottom Plates.*—With these—

$$s = \sqrt{\frac{3 \cdot p}{800\text{K}}}\left[d - r\left(1 + \frac{2r}{d}\right)\right]$$

Here s, p, and K have the same meaning as before.

r = inner radius of rounding of dog stay in millimetres.

d = inner diameter of the bottom plate in millimetres.

4. Rounded Thick Bottom Plates without Staying.

The thickness of these (to resist internal pressure) is—

$$s = \frac{p \cdot r}{200k}$$

Here s and k denote the same as before.

r = radius of the arc of the circle in millimetres, assuming that it is about equal to the diameter of the corresponding boiler shell.

k = allowable stress on the material in kilogrammes per square millimetre, namely—

For wrought iron up to 4·5 kilogrammes.
For mild steel up to 6·0 kilogrammes.
For copper up to 2·5 kilogrammes.

5. Stays and Screwed Stays.

The stress in these should not exceed—

5 kilogrammes per square millimetre for unwelded iron stays.
6 „ „ „ „ mild steel stays.
3 „ „ „ copper stays.

If the pressure is high (10 atmospheres and above) it is advisable to screw those longitudinal stays which have nuts, as well as the stay tubes, into the plates they support, and the former should also be fitted inside and out with nuts, a suitable washer being fitted under each outer nut; the stay tubes should, however, be expanded and beaded over. The end stays should be as long as possible.

6. Stays for Fire-box Tops.

The projecting unattached girders must be calculated as follows:—

$$b = \frac{p(w-d)e\,l}{ch^2}$$

Here p = maximum working pressure in atmospheres.
w = width of fire box in millimetres.
d = distance of stay bolts apart in millimetres.
e = distance of girders apart in millimetres.
l = length of girder in millimetres.
h = depth of girder in millimetres.
b = width of girder in millimetres (or total thickness of girder plates).
c = 420 if there is one stay to each girder.
c = 630 if there are two or three stays to each girder.
c = 720 if there are more than three stays to each girder.

If the girders supporting the top are suspended, they must be calculated according to the altered proportions of the load upon them.

§ 299. Extract from Rules of the "Bureau Veritas."

(*a*.) The thickness of the shell plates and rivets is calculated from the following formulæ—

$$(1.) \qquad t = \frac{\text{P} \cdot \text{D}}{20a\text{R}} + 1.$$

Here t = thickness of shell plate in millimetres.
P = working pressure above atmosphere in atmospheres.
D = greatest inside diameter of boiler shell in centimetres.
R = allowable tensile stress, kilogrammes per square millimetre.

The latter is—

For iron or steel $\text{R} = \frac{\text{minimum tensile strength of material}}{4}$

For iron R = 7·9 kilogrammes per square millimetre, if the minimum tensile strength of material is not known.

Further $a = \frac{p-d}{p}$

p = pitch of rivets in outer row in millimetres.
d = diameter of rivet holes in millimetres

$$(2.) \qquad A = \frac{P \cdot D \cdot l}{2s}$$

Here P and D are the same as before.

l = pitch of rivets in the outer rows in centimetres.

S = maximum stress in pounds per square inch which will be allowed on the rivets.

For steel rivets, one-fifth part of the ultimate tensile strength can be taken.

For iron rivets, 6·3 kilogrammes per square millimetre, corresponding to an ultimate tensile strength of about 31·5 kilogrammes per square millimetre.

A = total shearing surface in square inches of the rivets (*i.e.*, twice the area of the rivet hole when the rivet is in double shear), when machine riveted.

Only $\frac{15}{16}$ of the full area must be taken when the riveting is done by hand.

$$(3.) \qquad B = \frac{P \cdot D \cdot l}{2R} - \frac{C \cdot S}{R}$$

Here P, D, l, R, and S have the same equivalents as before.

B = sectional area in square millimetres of the plate, on portion l of the joint, along the line of its supposed rupture, assuming thickness of plate to be reduced by 1 millimetre, due to corrosion.

C = total sectional area of rivets which are exposed to shear in the length l.

$$(4.) \qquad t = \frac{PD}{20aR} + 1$$

$$a = \frac{q - d}{q}$$

Here t = thickness in millimetres of single butt strap, or sum of thicknesses, if there are two straps.

q = pitch of rivets in inner row in millimetres.

d = diameter in millimetres of rivet holes in inner row.

Remarks.—If the above formulæ are used, it is assumed that all the rivet holes are drilled after the plates are bent. These holes must be at the distance of the diameter of *one* rivet from the edge of the plate. In zigzag riveting, the distance between the rows is to be such that there is no fear of a rupture through plate or butt strap along the zigzag line. When stays are bolted through the shell, they should be so arranged that they do not weaken the shell plates more than the

riveted joints. For circumferential seams, double riveting will be required if the thickness of the plates exceeds $12\frac{1}{2}$ millimetres ($\frac{1}{2}$ inch). In double-ended boilers with six furnaces, treble riveting will be required for the circumferential seams connecting the shell rings with each other; it is not required for the end seams.

(*b.*) *Flat Plates.*—For these $t = 1{\cdot}5 + \sqrt{(a^2 + b^2)\frac{P \cdot K}{T}}$

Here P = working pressure above atmosphere in kilogrammes per square centimetre.

t = thickness of plate in millimetres.

a = pitch of stays in one row in centimetres.

b = distance in centimetres between two rows of stays. In case of irregular staying, the mean distance between the stays is to be substituted for $\sqrt{a^2 + b^2}$.

T = tensile strength of material of plates in kilogrammes per square millimetre.

K = 0·735 when the stays are screwed into the plates and riveted over.

K = 0·578 when the stays are screwed into the plates, and fitted with outside nuts at either end.

K = 0·542 when the stays are screwed into the plates, and fitted with nuts and washers inside and out, and the outer washer has a diameter of at least $0{\cdot}4a$, and a thickness of at least $\frac{2}{3}t$.

K = 0·481 when the stays are fitted with inside and outside nuts and washers, the outside washer being riveted to the plate, and at least $0{\cdot}75t$ thick, and its diameter at least $0{\cdot}6a$.

In flat plates which are in contact on one side with the steam, on the other with the hot gases—

$$t = 3 + \sqrt{(a^2 + b^2)\frac{P \cdot K}{0{\cdot}9T}}$$

When the front plates are in two parts, the lap joints must be double riveted if the plate is 13 millimetres ($\frac{1}{2}$ inch) or above.

(*c.*) *Stays and Screw Stays.*—For these—

$$d = 3 \text{ millimetres} + \sqrt{\frac{7{\cdot}5Q}{T}}$$

Where d = inner diameter of the stay in millimetres.

Q = total load upon stay in kilogrammes.

T = tensile strength of the material in kilogrammes per square millimetre, namely—

For steel, the lower limit assumed for tensile strength (tensile strength, 35 to 47 kilogrammes per square millimetre).

For iron, 35 kilogrammes per square millimetre.

The stay tubes must be screwed into the plates they support.

(*d.*) *Cylindrical Furnaces.*

(1.) The thickness of plain cylindrical furnaces should be—

$$t = \sqrt{\frac{P \cdot D \cdot L}{C}} \text{ for iron.}$$

$$t = \sqrt{\frac{P \cdot D \cdot L}{1{\cdot}2C}} \text{ for mild steel.}$$

t = required thickness of plate in millimetres.

D = outside diameter of furnace in centimetres.

P = working pressure in kilogrammes per square centimetre above atmosphere.

L = length of furnace in centimetres, or if made with efficient rings the length between the rings.

C = 588 when the furnace is truly circular, and the longitudinal seams are welded and butt-jointed, or lapped, bevelled, and double riveted.

The thickness of tube must not, however, be less than—

$$t = \frac{P \cdot D}{56} \text{ for iron plates.}$$

$$t = \frac{P \cdot D}{63} \text{ for steel plates.}$$

(2.) With *corrugated* furnaces—

$$t = \frac{P \cdot D}{139} + 3.$$

P and t having the same meaning as before.

D = outside diameter in centimetres measured across the top of the corrugations.

It is here assumed that the

Depth of the corrugations is at least 4 centimetres (1½ inches).

Length „ „ 15 „ (6 „).

(3.) With *ribbed* furnaces—

$$t = \frac{P \cdot D}{130} + 3.$$

D denoting the greatest outside diameter between the ribs in centimetres.

Here it is assumed that the distance between the ribs = 23 cm.
Height of the ribs = 35 mm.

It is assumed that the material used has a tensile strength of 41 to 47 kilogrammes per square millimetre (26 to 30 tons per square inch).

§ 300. Extract from Lloyd's "Regulations for British and Foreign Shipping."

1. *Boiler Shells.* $t = \frac{A \cdot D}{C \cdot B}$ for iron boilers.

$$T = \frac{A \cdot D}{C \cdot B} + 2 \text{ for steel boilers.}$$

Where t = thickness of plate in inches.
T = thickness of plate in sixteenths of an inch.
A = working pressure in pounds per square inch.
D = mean diameter of shell in inches.
B = percentage of strength of joint.
C = coefficient according to following table.

Values of C *for Iron Boiler Shells* (Lloyd's Rules).

Description of Joint.	For Plates ½ inch thick and under.	For Plates ⅝ inch thick and above ½ inch.	For Plates above ¾ inch thick.
Lap joint—			
Punched holes - -	155	165	170
Drilled holes - -	170	180	190
Double butt strap joint—			
Punched holes - -	170	180	190
Drilled holes - -	180	190	200

Values of C *for Steel Boilers.*

C = 21 with double butt straps of equal width.
C = 20·25 with double butt straps, the outside strap having one row of rivets more than the inside.
C = 19·5 with lap joints.

If the tensile strength of the boiler shell plate is more than 27 tons per square inch, C may be correspondingly increased.

The inside butt strap must be at least $\frac{3}{4}$ the strength of the longitudinal joint.

B = the minimum percentage of strength of the longitudinal joint for plate at joint, found as follows:—

$$B = \frac{p - d}{p} \times 100;$$

and for the rivets at joint

$$B = \frac{n \cdot a}{p \cdot t} \cdot 90 \text{ for iron plates and iron rivets, drilled holes.}$$

$$B = \frac{n \cdot a}{p \cdot t} \cdot 85 \text{ „ steel „ steel „}$$

$$B = \frac{n \cdot a}{p \cdot t} \cdot 70 \text{ „ steel „ iron „}$$

Where p = pitch of the rivets in inches.
t = thickness of plate in inches.
d = diameter of rivet holes in inches.
n = number of rivets used per pitch in a longitudinal joint.
a = sectional area of rivet in square inches.

In case of rivets in double shear $1{\cdot}75a$ is to be used instead of a.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

Shell plates under domes in boilers so fitted, to be stayed from the top of the dome, or otherwise stiffened.

Note.—For the shell plates of superheaters or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the coefficients in the above tables should be $\frac{2}{3}$ of those given.

Allowable Strains upon Stays, Stay Bolts, and Stay Tubes in pounds per square inch (Lloyd's Rules).

Description of Stay.	Diameter at Bottom of Thread less than 1½ inch.	Diameter at Bottom of Thread more than 1½ inch.
Iron stays - - -	6,000 lb. per sq. in. (welded).	7,500 lb. per sq. in. (unwelded).
Steel stays (screw stays)*	8,000 lb. per sq. in.	9,000 lb. per sq. in.
Steel stays (other stays)*	9,000 „ „	10,000 „ „

* No steel stays are to be welded.

The maximum stress in stay tubes must not exceed 7,500 lb. per square inch.

2. *Flat Plates or Stayed Surfaces.*—For these $T = P\sqrt{\frac{A}{C}}$ or $A = \frac{C \times T^2}{P^2}$

Where P = pitch of stays in inches. If the pitch in the rows is not equal to that between the rows, the mean of the squares of the two pitches is to be taken.

T and A as before.

C = constant, the values of which are given in the table on next page.

Where doubling plates are employed and securely riveted to the flat plates, their thickness t in sixteenths of an inch is $t = 2\left(P\sqrt{\frac{A}{C}} - T\right)$ or $A = \frac{C \times \left(T + \frac{t}{2}\right)^2}{P^2}$ where t is not to be less than two-thirds of T.

In the case of front plates of boilers in the steam space, C is to be taken at 20 % less, except where the plates are shielded from the direct action of the heat.

For the wide water spaces between the nests of tubes $T = P\sqrt{\frac{A}{C}}$ or $A = \frac{C \times T^2}{P^2}$. Here P = horizontal distance from centre to centre of the bounding rows of tubes, and C as follows:—

C = 120, if every third tube is a stay tube, and not fitted with nuts outside the plates.

C = 130, if every third tube is a stay tube, but with nuts outside the plates.

C = 140, if every second tube is a stay tube, and not fitted with nuts outside the plates.

C = 150, if every second tube is a stay tube, with nuts outside the plates.

C = 160, if every tube is a stay tube, and not fitted with nuts outside the plates.

For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = A \quad \text{or} \quad T = P\sqrt{\frac{A}{140}}$$

where T and A are as before, and P = the mean pitch of stay tubes from centre to centre.

For the steel tubes in the vicinity of the nest of boiler tubes C must be taken as = 140; P is the mean pitch of the stay tubes.

Values of C *for Flat Plates* (Lloyd's Rules).

Kind of Plate.	Kind of Stays or Stay Bolts.	Thickness of Plates.	C.
Iron or steel -	Screw stays with riveted heads.	$\frac{7}{16}$ in. and under.	90
Do. -	Do.	Above $\frac{7}{16}$ inch.	100
Do. -	Screw stays with nuts.	$\frac{7}{16}$ in. and under.	110
Iron - - -	Do.	Above $\frac{7}{16}$ inch.	120
Steel - -	Do.	Above $\frac{7}{16}$ and under $\frac{9}{16}$ inch.	120
Do. - - -	Do.	$\frac{9}{16}$ in. and above.	135
Iron - - -	Screw stays with double nuts.	Do.	140
Steel not exposed to the fire -	Do.	Do.	175
Iron - -	Stays with double nuts, and washers outside the plates, having a diameter $= \frac{P}{3}$ and thickness $= \frac{T}{2}$.	Do.	150
Steel not exposed to the fire -	Do.	Do.	185
Iron - -	Stays with double nuts, and washers riveted to the outside of the plates, having a diameter $= 0{\cdot}4P$ and thickness $= \frac{T}{2}$.	Do.	160
Steel not exposed to the fire -	Do.	Do.	200
Iron - -	Stays as above, but diameter of washers $= \frac{2}{3}P$ and thickness $= T$.	Do.	175*
Steel not exposed to the fire -	Do.	Do.	220

The thickness of the tube plates of combustion chambers, in cases

* Or 190 if P be taken as the pitch of the stays in the row.

where the pressure on the top of the chamber is borne by these plates, is not to be less than that given by the following rule—

$$T = \frac{A \cdot W \cdot D}{1{,}750\ (D - d)}$$

Where A = working pressure in pounds per square inch.
W = width of combustion chamber over plates in inches.
D = horizontal pitch of tubes in inches.
T = thickness of tube plates in sixteenths of an inch.
d = inside diameter of plain tubes in inches.

3. The thickness of the *girders supporting the top of the combustion chamber* is taken at—

$$T = \frac{A\ (L - P)\ D \cdot L}{C \cdot d^2} \text{ or } A = \frac{C \times d^2 \times T}{(L - P) \times D \times L}$$

Where

L = width of combustion chamber between the plates
T = thickness of girders at centre
d = depth of girder at centre
D = distance from centre to centre of girders
P = pitch of stays in girder
A = working pressure in pounds per square inch
} inches.

C = 6,600 for steel girders with one stay to each girder.
C = 9,900 „ „ two or three stays to each girder.
C = 11,000 „ „ four or five „ „
C = 11,550 „ „ six or seven „ „
C = 11,880 „ „ eight or more „ „

4. *Furnaces.*—The strength of plain furnaces, where $L > 120T$, is

$$T = \sqrt{\frac{A \cdot L \cdot D}{1{,}075{,}200}} \text{ or } A = \frac{1{,}075{,}200 \times T^2}{L \times D}$$

Here A = working pressure in pounds per square inch.
T = thickness of plates in inches.
L = length of the plain cylindrical part in inches, measured from the centre of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the beginning of the curvature of the flanges of the furnace, where it is flanged or fitted with Adamson rings.
D = outside diameter of furnace in inches.

If the length of the plain part of the furnace is less than 120 times the thickness of the plate, then—

$$T = \frac{A \cdot D + 50L}{15{,}000} \text{ or } A = \frac{50 \times (300T - L)}{D}$$

If the above formulæ are employed, it is assumed that the steel will have a tensile strength of not less than 26, and not more than 30 tons per square inch. If the material of furnaces has a less strength than 26 tons per square inch, then for each ton per square inch below the minimum tensile strength of 26, the coefficient is to be correspondingly decreased by $\frac{1}{26}$th part.

With steel corrugated tubes $T = \frac{A \cdot D}{C} + 2$, or $A = \frac{C \times (T - 2)}{D}$

T being = thickness of plate in sixteenths of an inch.
D ,, = outside diameter of corrugated furnaces, or smallest outside diameter of ribbed furnaces in inches.
C ,, = 1259 with Fox, Morrison, Deighton, or Beardmore's corrugated furnaces.
C ,, = 1160 with ribbed furnaces (spacing between the ribs 9 inches).
C ,, = 912 with spirally corrugated furnaces.
C ,, = 945 with Holmes' patent furnaces, corrugation not more than 16 inches apart centre to centre, and not less than 2 inches high (T for plain portions, and D for plain parts).

TABLE

Cylindrical

	Type of Ship	Fast steamer	Fast steamer	Cargo steamer	Cargo steamer
	Kind of Draught used	Natural	Howden	Natural	Natural
	Type of Boiler	Double ended	Double ended	Double ended	Single ended
	Heating Surface *sq. ft.*	6,466	6,090	3,980	2,000
	Grate Area *sq. ft.*	200	155	110	55
	Heating Surface : Grate Area *ratio*	32·1	39·3	36·3	36·4
	Press. above Atmosphere *lbs. per sq. in.*	178	214	214	214
Boiler Shell	Internal Diameter *inches*	200	196	162	162
Boiler Shell	Length *inches*	244	244	238	132
Boiler Shell	Thickness of Shell Plate *inches*	$1\frac{7}{16}$	$1\frac{11}{32}$	$1\frac{7}{16}$	$1\frac{7}{16}$
Boiler Shell	Tensile Strength of material *tons per sq. in.*	28—30	33—36	28—32	28—32
Furnace	Type of Furnace	Purves	Morison	Morison	Morison
Furnace	Number of Furnaces	8	8	6	3
Furnace	Internal Diameter *inches*	$47\frac{1}{4}$	43·5	39·5	39·5
Furnace	Thickness of Furnace Plate *inches*	$\frac{19}{32}$	$\frac{23}{32}$	$\frac{21}{32}$	$\frac{21}{32}$
Furnace	Number of Combustion Chambers per boiler	3	3	3	3
Furnace	Minimum clear Space between Boiler Shell and Furnace *inches*	4·84	5	5·86	4·44
Boiler and Stay Tubes	Outside Diameter of Boiler Tubes and Stay Tubes *inches*	$3\frac{1}{4}$—$3\frac{1}{7}$	$2\frac{8}{4}$	3	3
Boiler and Stay Tubes	Thickness of Boiler Tubes (welded iron tubes) *inches*	0·16	0·13	0·15	0·15
Boiler and Stay Tubes	Thickness of Stay Tubes (welded iron tubes) *inches*	0·31	0·19-0·31	0·39	0·39
Boiler and Stay Tubes	Number of Boiler Tubes	674	592	352	171
Boiler and Stay Tubes	Number of Stay Tubes	194	364	222	111
Boiler and Stay Tubes	Length between the Tube Plates *inches*	$92\frac{1}{2}$	94	94	92
Boiler and Stay Tubes	Vertical Pitch *inches*	4·16	3·85	4·09	4·09
Boiler and Stay Tubes	Horizontal Pitch *inches*	4·13	3·85	4·09	4·09
Boiler and Stay Tubes	Thickness of Tube Plates *inches*	0·94	0·94	0·98	0·98
Boiler and Stay Tubes	Lowest Water-Level above middle of Boiler *inches*	52·16	52·08	42·44	42·44
Boiler and Stay Tubes	Height of Combustion Chamber *inches*	49·36	53·38	47·2	26·77

No. 62.

Boilers.

Scout	Cruiser	Cargo steamer	Cargo steamer	Cargo steamer	Cargo steamer	Cargo steamer	Cargo steamer	Tug
Closed stokehold	Closed stokehold	Natural	Natural	Natural	Natural	Natural	Natural	Natural
Double ended	Double ended	Single ended	Single ended	Double ended	Single ended	Single ended	Single ended	Single ended
4,800	4,067	2,582	1,400	3,850	1,130	970	753	430
172	122	74	43	122	37	32	26	14
27·9	33·2	35·1	32·1	31·4	31·1	30	29	30·8
170	185	206	170	185	114	170	150	140
180	150	183	130	163	126	126	96	83
200	214	140	133	224	124	126	106	95
$1\frac{7}{32}$	$\frac{11}{16}$	$1\frac{1}{2}$	$\frac{11}{16}$	$1\frac{7}{32}$	$\frac{3}{4}$	1	$\frac{3}{4}$	$\frac{11}{16}$
—	—	28—32	28—32	28—32	28—32	26—29	26—29	26—29
Purves	Morison	Morison	Morison	Morison	Plain	Plain	Plain	Plain
8	6	3	2	2×3	2	2	2	1
39·5	37·5	$47\frac{1}{4}$	39·5	39·5	37	36	27·5	33·5
$\frac{17}{32}$	$\frac{5}{8}$	$\frac{33}{32}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{19}{32}$	$\frac{1}{2}$	$\frac{9}{16}$
2×2	3	3	2	3	2	2	2	1
3·58	3·74	5	3·46	4·72	5·70	—	—	—
3	$2\frac{1}{2}$	3	3	3	3	$3\frac{1}{4}$	3	3
0·13	0·11	0·15	0·15	0·15	0·13	—	—	—
0·23	0·23	0·39	0·39	0·31	0·39	—	—	—
580	618	251	144	430	132	84	80	62
136	190	125	62	164	40	54	40	18
$88\frac{1}{2}$	83	90	$88\frac{1}{2}$	$86\frac{1}{2}$	86	83	79	67
3·85	3·30	3·93	3·93	4	3·97	4·48	3·85	3·9
3·85	3·38	3·93	3·89	4	3·93	4·48	3·85	3·9
0·86	0·90	0·96	0·82	0·86	0·70	0·92	0·70	0·70
45·27	40·63	46·18	35·43	40·78	32	—	—	—
22·04	45·43	26·77	24·8	49·60	20·47	—	—	—

SECTION III.

LOCOMOTIVE BOILERS.

§ 301. **Dimensions of Locomotive Boilers.** — Locomotive boilers were formerly in use for all light war vessels. Since the beginning of the last decade they have been almost entirely superseded by water-tube boilers, and are now scarcely ever constructed, except to replace old boilers of this type. For method of construction see Figs. 447, 448.

In newer boilers of this type the lower edge of the fire box, which is attached to the boiler shell, forms the lower part of the space dividing the fire box from the shell (in Figs. 447, 448 the division is made by a rectangular wrought-iron ring).

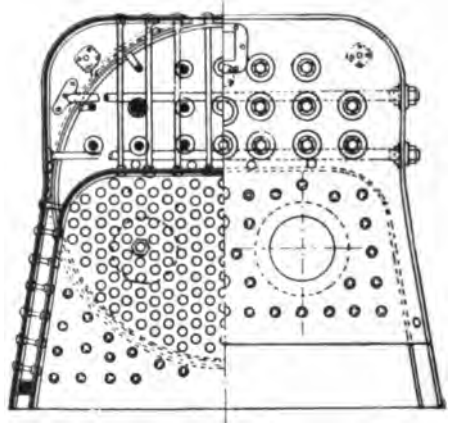


Fig. 447.

Dimensions, &c., of a modern marine locomotive boiler. Power developed, 900 I.H.P. with forced, 600 I.H.P. with natural draught. Pressure above atmosphere, about 160 lb. per square inch. Heating surface, 1,700 square feet. Grate area, 470 square feet. Total length, 19 feet. Diameter of the shell (cylindrical part containing the tubes) = 7 feet 9 inches. Length of the tubes, 7 feet $1\frac{1}{2}$ inch. External diameter of the tubes, 2 inches. Number of tubes, 420, of which 90 are stay tubes. Width of the furnace (rectangular part containing the grate), 8 feet 6 inches at bottom, 7 feet 9 inches at top. Height, 8 feet 2 inches, of which 4 feet 2 inches are below the middle of the shell. Width of grate = 2 feet × 3 feet 6 inches. Length of grate = 6 feet 11 inches.

When used in light war vessels, these locomotive boilers are generally highly forced (pressure of air in the combustion chamber or stokehold

being as much as $4\frac{1}{2}$ to 5 inches, and the fuel burnt per square foot of grate surface per hour as much as 82 lbs.

The ratio of heating surface to grate area varies considerably, from $\frac{H}{R} = 35$ to 55. See also § 321, and Table No. 68.

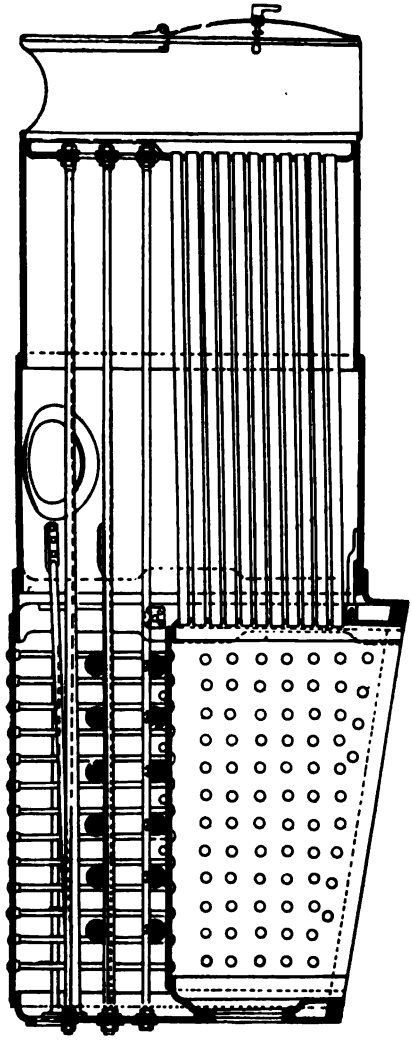


Fig. 448.

SECTION IV.

WATER-TUBE BOILERS.

§ 302. **General Remarks.**—Water-tube boilers are now almost always fitted in light war vessels, torpedo-boats and destroyers, and cruisers. In the larger class of warships they are frequently fitted conjointly with cylindrical boilers. Hitherto they have been but little adopted in the mercantile marine.

The advantages of water-tube boilers may be stated in general to be *—

1. Capacity to raise steam rapidly.
2. Facility in responding to a change of speed in the engines.
3. Facility with which they can be fitted on board ship.
4. Less danger in the event of an explosion.
5. Capacity for being greatly forced.
6. Reduced weight.

The disadvantages are—

1. Increased susceptibility to irregular feeding.
2. Increased susceptibility to irregular stoking.
3. Increased susceptibility to rapid fouling.
4. Difficulty of internal cleaning.

A distinction is made between water-tube boilers with large and with small tubes. The Belleville, Dürr, and Niclausse boilers belong to the first class. To the second class belong the Yarrow, Normand, Thornycroft, Schulz, and other similar boilers, none of which (except the Yarrow) differ from the general type by any very marked characteristics. A description is given below of some typical examples of water-tube boilers. A few general remarks, which apply to the

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* Compare the remarkable essay in *Marine Rundschau*, 1901, page 524, "The Water-tube Boiler Question in the German Navy," by Marine Oberbaurat Köhn von Jaski; also *Schiffbau*, 1901, iii., page 129 *et seq.*, "Further Contributions to the Water-tube Boiler Question in the German Navy," by Züblin.

construction of them all, will be found in the description of the different systems.

For dimensions of grate surface, see Table No. 59, page 469.

For calculation of the heating surface, see Tables of Details of Boilers in the description of the different systems.

For flue area in the nests of boiler tubes, see page 539.

For circulation of water and sectional area of the headers, see page 548.

For securing the brickwork for the furnaces, see page 539.

For fitting the tubes into the upper and lower drums, see pages 526 and 539.

For regulating the feed-water, see page 516.

For zinc plates, see page 526.

Material for Water-tube Boilers.—The boiler plates are made of the same material as those of cylindrical boilers, viz., Siemens-Martin steel of about 27 to 32 tons per square inch tensile strength for plates which are not welded or in contact with the fire, and 23 to 27 tons for those exposed to the fire. Elongation of the former, at least 20 per cent.; of the latter, at least 25 per cent. Stress upon the plates about 8,500 pounds per square inch.

Tubes.—These are always seamless or solid drawn, and made of soft iron or Siemens-Martin steel. They are either drawn without a seam, or, in the latest practice, are rolled while hot, without a seam, according to the Mannesmann process, and without being further drawn. The latter class of tubes are as durable as the seamless drawn tubes.

§ 303. **Belleville Boiler.**—This boiler consists in the main of vertical parallel sets of tubes known as "elements," arranged side by side, which are connected at their lower end with a feed collector, and at their upper end to a steam drum, both being placed at right angles to the direction of the tubes (see Figs. 449 to 454). The elements consist of straight solid drawn steel tubes (from about 6 feet 6 inches to 7 feet long, and 4 to 4½ inches diameter, ·2 inch thick) arranged by means of so-called "junction boxes" to form zigzags. Each tube is inclined about 4° to the horizontal. The ends of the tubes are fitted into junction boxes of cast steel or malleable cast iron, about ·3 inch thick. They are screwed into the back junction boxes with a fine thread, and connected to the front junction boxes by a short pipe and socket, forming a sleeve or muff coupling. The joint is made tight by means of screwed rings, which act as lock nuts (see Fig. 449). In each front junction box there are two small oval holes, closed by a door opening inwards, for

cleaning the tubes. The lowest junction box of each element has a conical opening, which is fitted with a nickel ring which forms a joint with a conical nipple secured to the feed collector. Each of the uppermost junction boxes is secured by means of a flange to a short length of vertical pipe screwed into the steam drum.

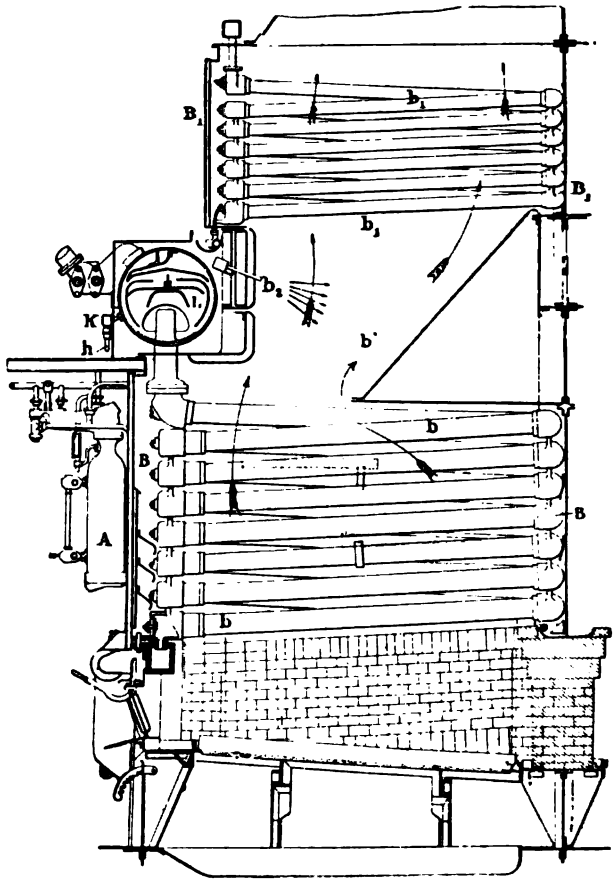


Fig. 449.

The feed collector is a horizontal rectangular steel tube about $4 \times 4 \times \frac{9}{16}$ inches thick. The pipes forming the connection to the lower row of junction boxes are screwed into the top of the collector side by side. The steam drum L lies above and parallel to the feed collector. It is a riveted, or welded, mild-steel or wrought-iron tube, about 20 inches internal diameter, and has cast-steel ends. Pipes

with flanges, to form the joints with the upper row of junction boxes, are screwed into the bottom of this drum. To separate the water carried over with the steam, a complicated system of baffles is provided inside the steam drum. The feed collector and steam drum lie in the same vertical plane as the front junction boxes. Between them, and at right angles to them, are fitted seven or eight vertical

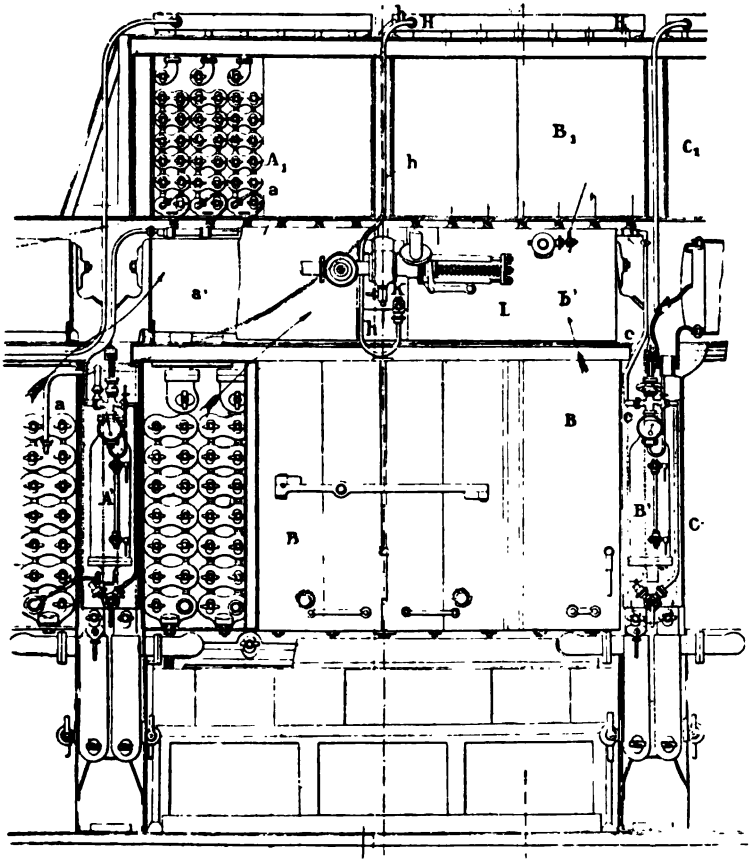


Fig. 450.

elements, each consisting of about twenty tubes if there is not, and about fourteen tubes if there is, an "economiser." The junction boxes are supported one on the other (see Fig. 450) by small lugs. The feed-water is sprayed into the steam drum (through a valve) and then flows down into cast-steel mud boxes, through two vertical pipes c, which are fitted to each steam drum. Here

it deposits any dirt, which can then be easily drawn off. The water passes thence to the feed collector, and from there into the tubes forming the elements. The lowest water level is situated about halfway up the elements. The lowest rows of tubes sometimes consist of "Serve" tubes, *i.e.*, tubes with internal ribs, a cross section of which is given at Fig. 451.

As the level of water in the elements is subject to great fluctuations, an *automatic feed-water regulator* (see Fig. 452) is fitted to all Belleville boilers. It consists of a tank 1, which is connected to one of the lower junction boxes through pipe 2, and to one of the upper junction boxes through pipe 3. A float 5 moves up and down in the tank. If the level of water falls, the float draws down lever 6, forces up lever 7 through rod 8, and opens valve 4 in the feed-water pipe by means of connecting rod 9. The arrangement is adjusted by weights 10 and spring 11, A being the fixed, and B the adjustable weights. Valves 12 and 13 serve to open connection with the feed-water pipes 14 and 15 respectively. On the outside of tank 1 are fixed the water gauge 17, pet-cocks 18, and bracket 19 for holding the regulator.

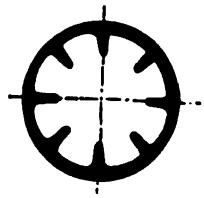


Fig. 451.

To utilise the heat in the escaping gases of combustion, which are at a fairly high temperature, and to further cool them, and prevent overheating of the uptake and funnel, a feed-water heater or *economiser* is placed above the boiler in the newer types (see Figs. 449 and 450, and the photograph Fig. 454). It is constructed in the same way as the boiler; internal diameter of the tubes = $2\frac{3}{4}$ inches. The junction boxes are malleable castings or cast steel. The rise of temperature of the feed-water in the economiser is 120° to 180°. The water passes from the automatic feed regulator A′B′ (Fig. 450) through pipe *c* to the feed-water collector of the economiser $A_1B_1C_1$, flows through the elements of the latter, and collects in the cross pipe H, which unites the upper ends of all the elements. Thence it passes through a pipe *h* and the feed valve K to the steam drum L. Between the boiler and economiser there is a combustion chamber *a′b′*, into which a current of air b_2 can be forced, so that the gases may be thoroughly mixed, and complete combustion ensured. Air is also sometimes forced through nozzles into the furnace immediately above the layer of coal on the grate, so that the smoke may as far as possible be consumed. The pressure of this air, which is compressed in separate air-compressors, is about 20 lb. per square inch.

In the Belleville boiler the steam is almost always generated at a pressure of about 40 lb. per square inch above the working pressure at the engines, and is reduced to the latter pressure immediately before

it enters the engine by means of a *reducing valve* (see Fig. 453). The steam enters the reducing valve from the boiler through the opening on the left, and, after passing through the slots, passes out through the opening on the right to the steam pipe and thence to the engine. At the same time it finds its way into the space above the valve through holes pierced in it. This exerts a pressure on the small piston, working in the stuffing box, and tends to close the valve. The upward pressure on this piston acts in a contrary direction to the

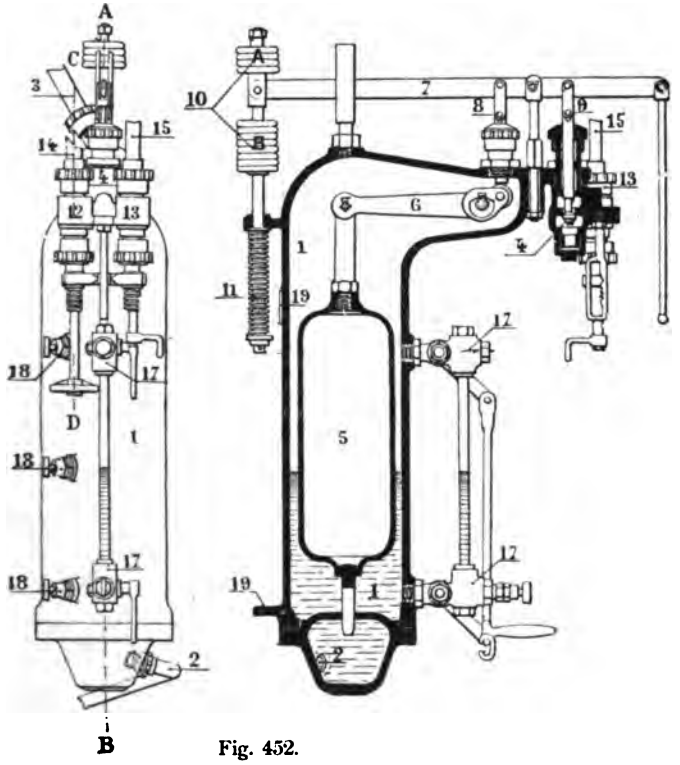


Fig. 452.

springs, which press down the piston through a single-armed lever, and tend to hold the valve open. If the required tension be given to the springs by means of the hand wheel and spindle shown on the left in Fig. 453, the throttling of the steam can be regulated at will. By using a higher pressure of steam in the boilers, a certain small reserve of energy is provided, which is an advantage, considering the small quantity of water and heat contained in the Belleville boiler. For the effect of throttling upon the dryness of the steam see page 472; for the dimen-

sions of boiler casing, grate, &c., see Figs. 449 and 454. The dimensions and efficiency of the Belleville boiler are given in Table No. 63. These boilers can be placed either athwart ship or fore and aft.

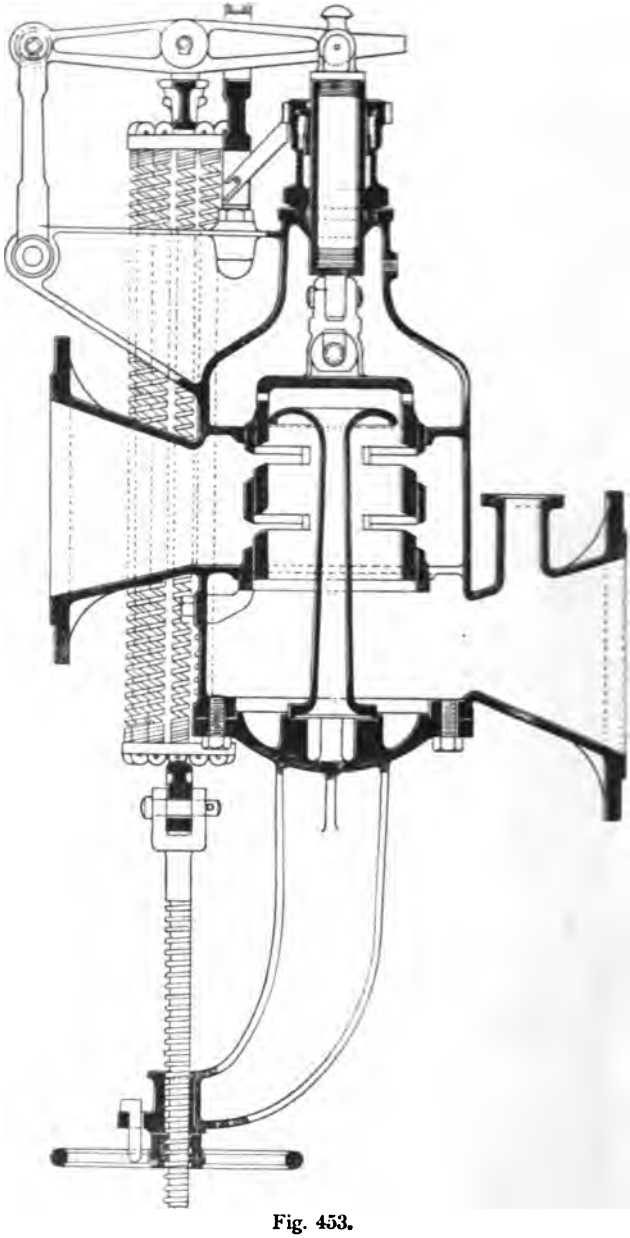


Fig. 453.

TABLE No. 63—*Belleville Boilers.*

Name of Ship.	I.H.P.* Maker's Figures.	Type of Engine.	No. of Boilers.	Steam Pressure. At the Engine.	Steam Pressure. In Boiler.	Total Heating Surface H.	Total Grate Area R.	Ratio H : R.	I.H.P. per sq. ft. of H.	I.H.P. per sq. ft. of R.	Consumption of Coal per I.H.P. per hour.	Consumption of Coal per sq. ft. of Grate Area per hour.	Air Pressure in Stokehold.	Authority.
				lb. per sq. in.		sq ft.					lb.	lb.	inches of water	
H.M.S. "Diadem," 1st Class Armoured Cruiser, 1898	17,190*	2 Engines, 4 Cylinders, Triple Expansion	30 with Economisers.	245	290	40,500 with Economiser	1440	28	0·42	12	·178	21	0·44	*Bulletin de l'Association Technique Maritime*, 1899, No. 10.
	12,800*			240	280	40,500 with Economiser	1440	28	0·31	9	·160	14·3	0·34	
H.M.S. "Niobe," 1st Class Armoured Cruiser, 1898	16,820*	2 Engines, 4 Cylinders, Triple Expansion	30 with Economisers.	235	288	40,500 with Economiser	1440	28	0·41	11	·165	19·3	0·28	Do.
	12,970*			220	256	40,500 with Economiser	1440	28	0·32	9	·156	14	0·20	
H.M.S. "Formidable," Battleship, 1899	15,000	2 Engines, 3 Cylinders, Triple Expansion	20 with Economisers.	250	300	37,000 with Economiser	1170	32	0·40	12·8	—	—	—	*Schiffbau*, iii., p. 178, and *Marine Rundschau*, 1901, p. 439.
	15,730*			—	- -	37,000 with Economiser	1170	32	0·42	13	·178	24	—	
"Gaulois," French Battleship, 1898	14,960*	2 Engines, 3 Cylinders, Triple Expansion	20	—	213	32,300	1076	about 30	about 0·46	about 14	·178	24·6	—	Details of the "Gaulois" and the 'Charlemagne" communicated by the builders, 1898.
	9,215*		—	—	—	32,300	1076	about 30	about 0·46	about 8·5	·149	13	—	
"Hertha" and "Hausa," large German Cruisers, 1897	10,000	3 Engines, 4 Cylinders, Triple Expansion	18 without Economisers.	185	256	25,000 without Economiser	795	31·5	0·39	12·5	—	—	—	*Schiffbau*, iii., p. 136.
				—	—	— without Economiser	—	—	—	—	—	—	—	
H.M.S. "Arrogant," 2nd Class Cruiser, 1897	10,340*	2 Engines, 3 Cylinders, Triple Expansion	18 without Economisers.	—	250	25,600 without Economiser	866	29·5	0·40	12	—	—	natural	*Bulletin de l'Association Technique Maritime*, 1899, No. 10.
	7,650*			—	247	25,600 without Economiser	866	29·5	0·30	8·8	·291	18·5	natural	

* The figures marked with an asterisk in this column are the results given on the trial trips.

§ 304. **Dürr Boiler** (see Figs. 455, 456).—This boiler consists of an upper drum, below which, and riveted to it, is a flat water space or header chamber. Straight parallel boiler tubes, closed at their further end, are fitted in the back plate of the water space. The water space is separated into two parts throughout its whole length by a dividing wall parallel to the tube plate. Into this wall small tubes are fixed, which are passed into the boiler tubes, and extend almost throughout their whole length.

The water level is rather above the bottom of the upper drum. The steam generated in the boiler tubes rises up in the back part of the water space between the dividing wall and the back tube plate; while in the front part the water descends, and finds its way into the inner circulating tubes, inside the boiler tubes.

The *water space* is seamless, having no riveted joints, the metal being merely welded together, and stiffened with stays and stay bolts between the holes for the tubes and the caps. The water space is widened at the top like a wedge to facilitate the circulation.

Both the *boiler tubes* and the back wall of the water space are slightly inclined, and are fixed into the back tube plate by forcing them into conical openings, the axes of which are not quite in line with the axes of the corresponding tubes. The diameter of the tubes is decreased at the back, where they fit loosely into a wrought-iron grating. The two outside vertical rows of tubes at either side of the boiler are bent round to the right and left, immediately behind the tube plate, so as to form a tube wall on both sides of the grate, enclosing the nest of tubes. The inner water-circulating tubes are fitted into the dividing wall of the water space with bell-mouthed ends, and are made of thin sheet iron.

The caps which serve to close the holes provided for the insertion of the tubes are introduced from within, and are pressed into the conical holes by means of a forged screw and collar, no packing being required (see Fig. 457). The back end of the tube is also made tight either by a conical cap on the inside, or a bronze lock nut on the outside; the latter arrangement makes a better joint, and is more quickly taken to pieces (Fig. 457). The *upper drum* is generally at right angles to the tubes, as shown in Fig. 455. This is made of mild steel plates riveted together, and stiffened with stays where it is cut away for the water space.

The *superheater* or steam dryer consists of several horizontal boiler tubes fixed into the upper drum, and lying in the uptake. The steam from the drum is caused to circulate through these tubes by means of the inner circulating pipes. The dried steam passes back through the outer tubes into a pipe in the steam drum, and thence to the main stop valve (see Fig. 455). The circulating pipes receive the wet steam from

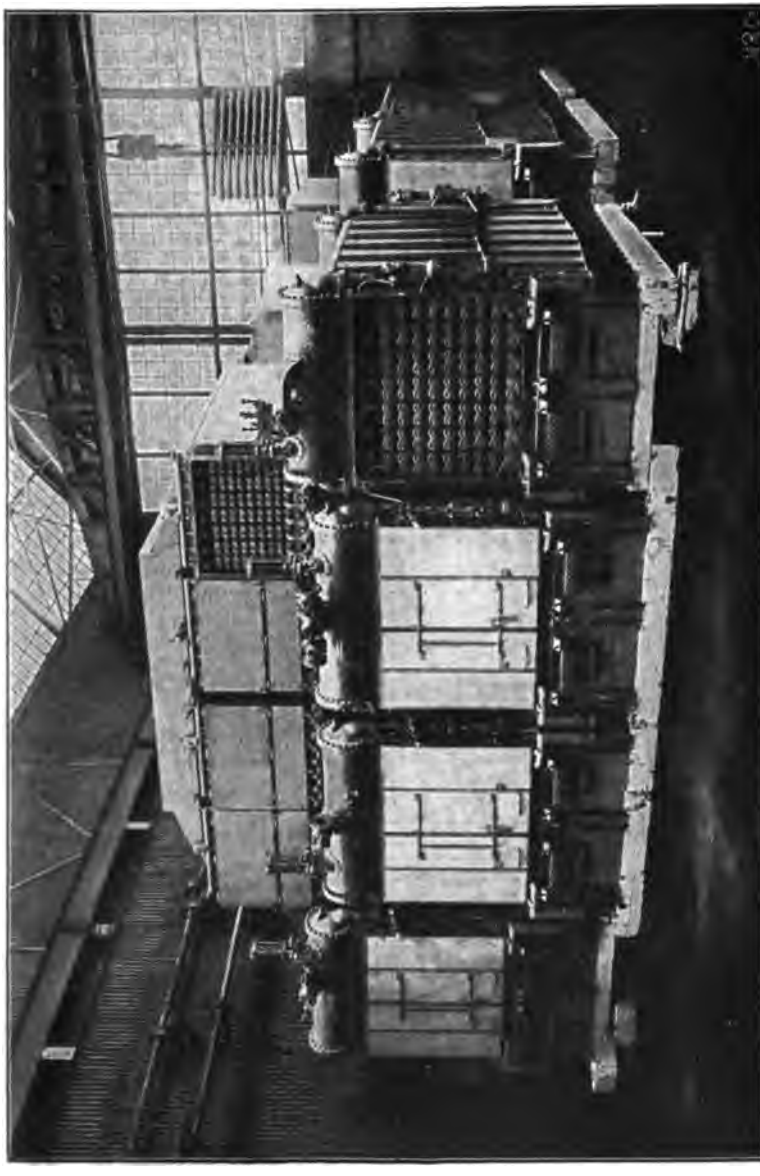


Fig. 454.

the upper drum through a pipe having small slots cut in it. The *grate* covers the whole area below the nest of tubes. The *combustion chamber* is lined with firebrick. Baffle plates are fixed above some of the rows of tubes, so that the flames are first forced backwards towards the back wall, then to the front, and then again backwards as

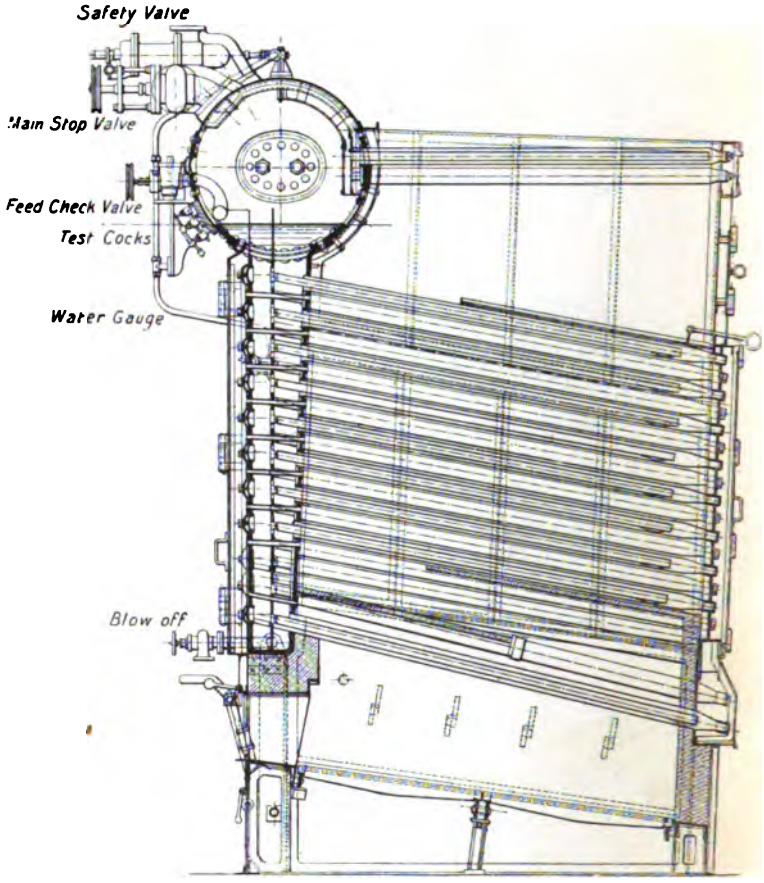


Fig. 455.

they rise through the nest of tubes. By means of iron rods passing through the holes for the stay bolts, these baffle plates can be shaken by the stoker and freed from the ashes which settle on them.

§ 304A. **Dimensions of a Dürr boiler** having 3,350 square feet

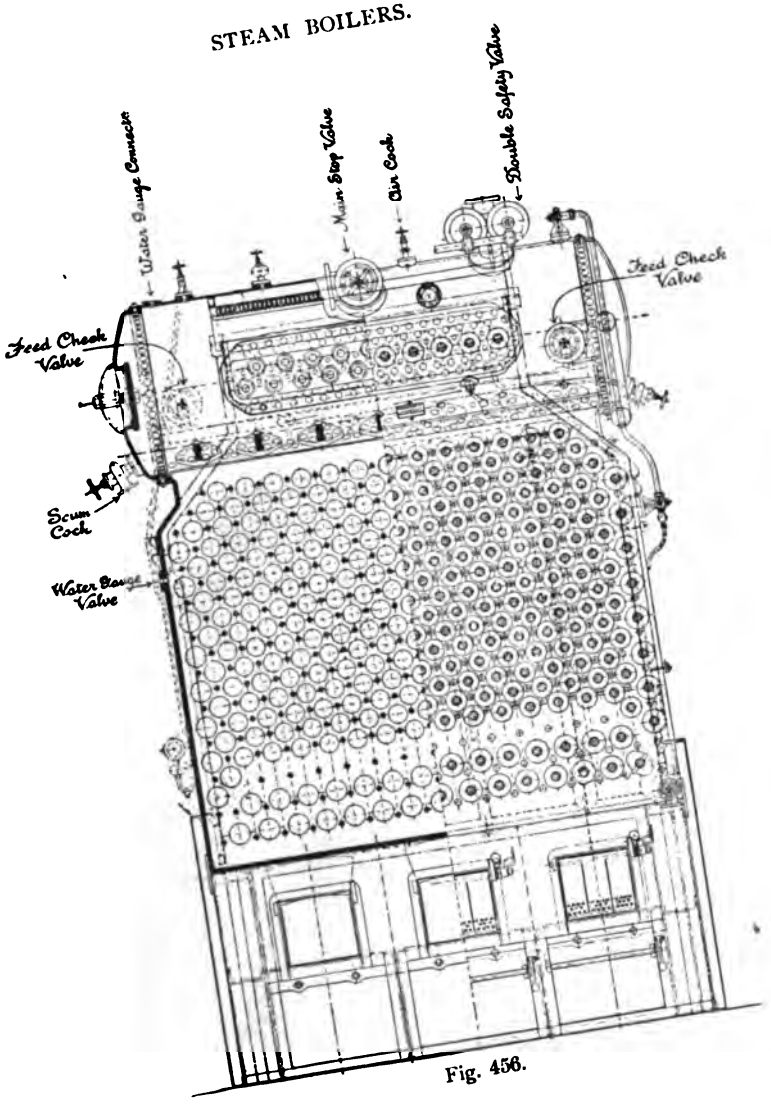


Fig. 456.

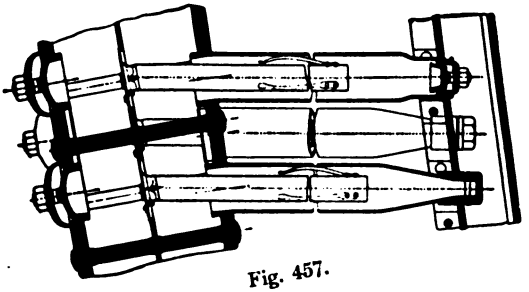


Fig. 457.

of wetted heating surface. Area of the superheater, 176 square feet; grate area, 80 square feet for a working pressure of 190 lb. per square inch.

Upper Drum.—Diameter, 45 inches; length over all, 13 feet 7 inches; thickness of plate, $\frac{5}{8}$ inch.

Water Space.—Width at top, 11·8 inches; at bottom, 7·8 inches; height, 7 feet 6 inches; breadth at top, 11 feet 5 inches; at bottom, 14 feet 10 inches; thickness of plate, $\frac{7}{8}$ to 1 inch.

Tubes.—20 rows: 16 rows at the top = 427 tubes; external diameter, $3\frac{1}{4}$ inches; thickness, $\frac{1}{8}$ inch. Next 2 rows = 56 tubes; external diameter, $3\frac{1}{4}$ inches; thickness, $\frac{5}{32}$ inch. Then 1 row = 28 tubes; external diameter, $3\frac{1}{2}$ inches; thickness of plate, $\frac{5}{32}$ inch. The lowest row = 28 tubes, has an external diameter of $3\frac{1}{2}$ inches; and thickness, ·2 inch. All the tubes are 7 feet 4 inches long. The tubes forming the water wall are $3\frac{1}{2}$ inches external diameter, ·128 to ·2 inch thick, 7 feet long. Tubes of the superheater = 44; external diameter, $2\frac{3}{4}$ inches; thickness of plate, ·138 inch; length, 6 feet.

Centre division plate of the water space, $\frac{1}{4}$ inch thick; distance from front tube plate, $4\frac{3}{8}$ inches.

Length of grate = 6 feet 6 inches: 6 firedoors, each 15 inches wide × 14 inches high.

Height of boiler over all, 15 feet; breadth of front, 15 feet 1 inch; depth of boiler, 8 feet.

Lowest water level, $13\frac{3}{4}$ inches below the centre of the upper drum.

Boiler Casing.—Thickness of inner plate, ·125 inch; of outer plate, ·04 inch; space between them, 2 to $2\frac{3}{4}$ inches. For dimensions and results of actual boilers see Table No. 64.

TABLE No. 64.—*Evaporative Experiments on Dürr Boilers in the German Navy.**

Name of Ship.	Dimensions of Boiler.			Pressure of Air in Stokehole.	Coal Consumption.		Working Pressure.	Water Evaporated.			Temperature in deg. F		
	Wetted Heating Surface.	Surface of Superheater.	Grate Area.		Per hour.	Per hour per square foot of Grate Area.		Per hour per square foot of Heating Surface.	Per pound of Coal.	Per pound of Coal reduced to Water at 32°, Steam at 212°.	In Chimney.	Steam in Superheater.	Degrees of Superheat.
	sq. ft.	sq. ft.	sq. ft.	inch.	lb.	lb.	lb. per sq. in.	lb.	lb.	lb.	deg. F.	deg. F.	deg. F.
"Victoria Louise"	2317·7	208·7	56·16	natural draught	990	17·67	213	4·10	9·59	9·37	536	—	—
"Sachsen" - -	2154·7	127·0	54·6	0·61	1564	28·67	186	5·78	7·96	8·09	734	392	10·6
"Vineta" - -	2167	118·3	50·3	0·98	1863	37·12	185	6·46	7·5	7·535	689	391	10·0
"Prinz Heinrich"	3057·3	168·2	72·63	natural draught	1113	15·36	192	3·35	9·166	9·125	538	—	—
Do.	3057·3	168·2	72·63	0·86	2258	31·18	206	5·94	8·035	8·227	599	—	—
Do.	3057·3	168·2	72·63	1·47	2684	37·04	192	6·64	7·545	7·711	611	—	—
"Prinz Adalbert"	3358·5	176·8	79·73	0·92	2640	33·2	189	6·30	8·06	8·003	591	—	—

* From data furnished by the Düsseldorf-Ratinger Water-tube Boiler Works, formerly Dürr & Co.

§ 305. **Yarrow Boiler** (see Figs. 459 to 461A).—This boiler consists of a riveted upper drum of Siemens-Martin steel, which is connected to two lower drums by straight tubes. In the larger boilers the lower drums are riveted, but in smaller boilers, where they cannot easily be examined internally, they are made in two halves and bolted together.

External diameter of tubes { In the older and smaller types, 1 to $1\frac{1}{8}$ inch. In the newer and larger types, up to $1\frac{3}{4}$ inch.

Thickness of Tubes:—For an external diameter of 1 inch, about $\frac{1}{10}$ inch for the inner rows of tubes, about $\frac{1}{16}$ inch for the outer; and $\frac{1}{8}$ inch with an external diameter of $1\frac{3}{4}$ inch. The tubes are secured to the upper and lower drums by expanding and coning the ends (see Fig. 458). To stiffen the tube plates of the lower drum a few tubes in the nest may be expanded into holes, the sides of which are slightly ribbed (see Fig. 467), in the same way as is done with bent tubes. They serve also to a certain extent as stay tubes.

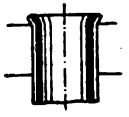


Fig. 458.

Dimensions.

Torpedo-boat (see page 41*).—One boiler. Heating surface, 1,420 square feet; grate area, 24·5 square feet; length over all, 9 feet 6 inches; breadth over all, 7 feet 9 inches; height over all, 5 feet; length of grate, 6 feet 3 inches; width, 4 feet; internal diameter of upper drum, 3 feet; thickness of metal, ·4 inch; external diameter of tubes, 1 inch; number of tubes, about 1,600; lower drum, 9 by 16 inches; pressure above atmosphere, 185 lb. per square inch. The tubes, which are seamless solid drawn steel, are expanded into the upper and lower drums. The tube plates where the tubes enter are made $1\frac{1}{8}$ inch thick for 1 inch tubes, and from $1\frac{3}{4}$ to 2 inches thick for $1\frac{3}{4}$ inch tubes. As is usual in most boilers of this class, the upper and lower drums are fitted with zinc plates. These must only be used in the upper drum with very great care, and in such a manner that, if eaten away, pieces of zinc cannot possibly fall off. If attention is not paid to this, and especially if, instead of rolled zinc plates, cast zinc plates are used, which fall to pieces very easily, one or more of the tubes may get stopped up, become red hot, and burst. There is usually no automatic feed-regulating valve, the feed-water being admitted to the lower part of the upper drum by means of an internal feed pipe provided with slots. In modern practice feed-water regulators, as described at page 542, are occasionally fitted.

* Compare *Schiffbau*, iii., No. 5, "Further Contributions to the Question of Water-tube Boilers," by Von Züblin.

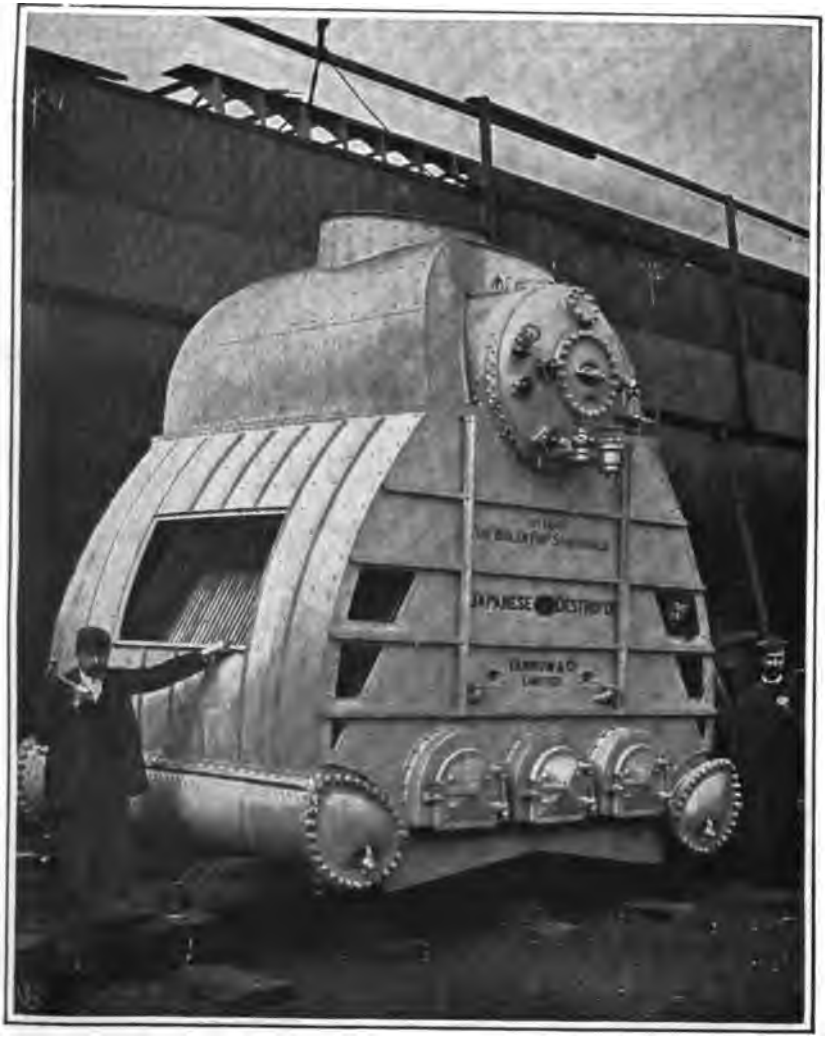


Fig. 459.

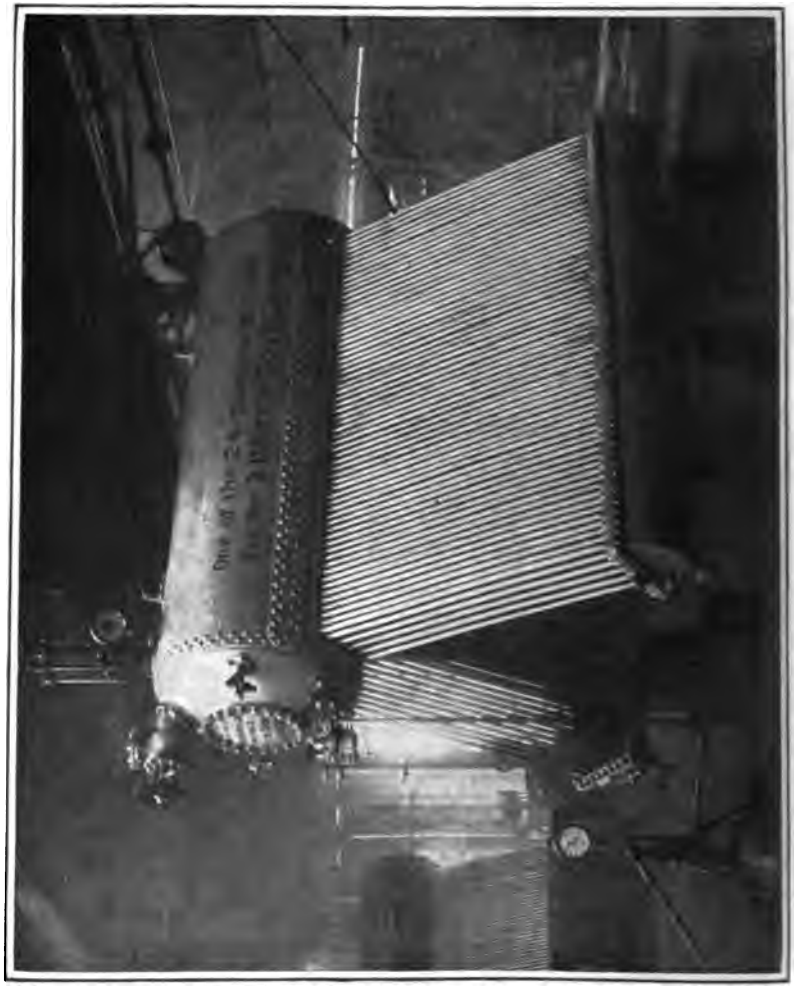


Fig. 140.

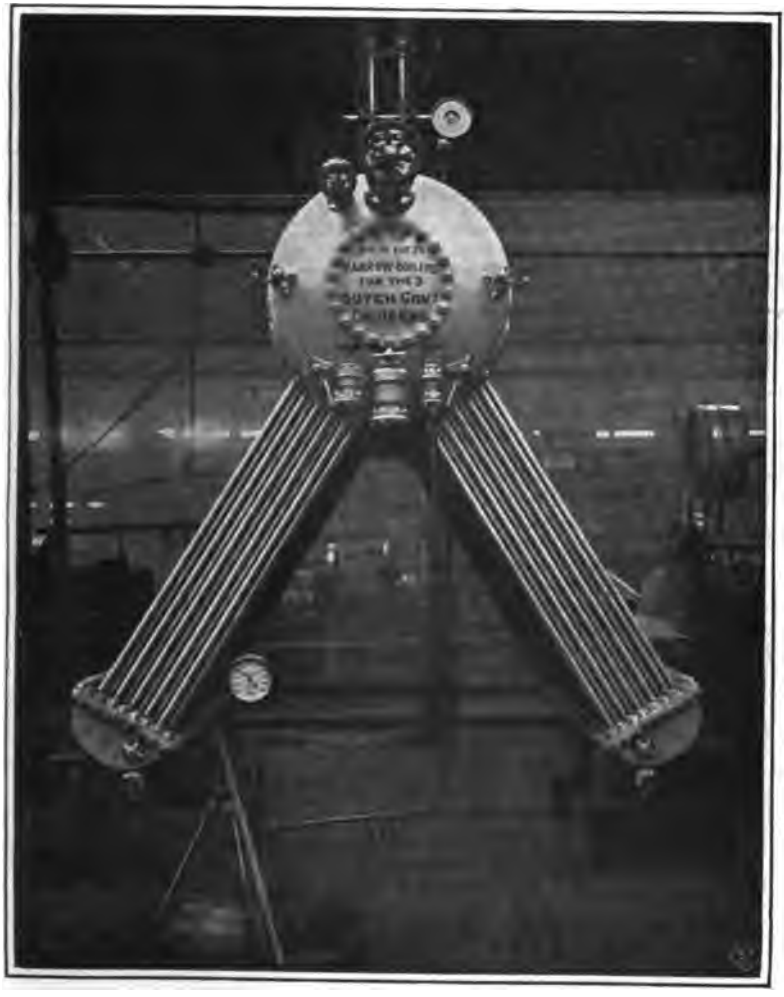


Fig. 461.

TABLE No. 65.— *Yarrow Boilers.*

Name of Ship.	I.H.P.* Maker's Figures.	Type of Engine.	Steam Pressure lb. per sq. in.		Total Heating Surface H.	Total Grate Area R.	Ratio H : R.	I.H.P. per sq. ft. H.	I.H.P. per sq. ft. R.	Consumption of Coal per I.H.P. per hour.	Air Pressure.	Authority.
			At the Engine.	In the Boiler.								
					sq. ft.	sq. ft.				lb.	in.	
"Viper," Austrian torpedo-boat, 1896; speed, 26·6 knots	1,940*	1 engine, triple expansion	—	174	3,733	70	53·4	0·52	27·8	—	0·86	*The Engineer*, April 3, 1896.
"Don Carlos I.," Portuguese cruiser	8,100	—	—	300	32,000	586	54·5	0·25	13·8	—	natural	*Engineering*, 1899, p. 185.
	12,670	—	—	—	32,000	586	54·5	0·40	21·5	—	forced	
"Zenta," Austrian cruiser, 1897	7,000	2 engines, 4 cylinder triple expansion	200	213	20,000	376	53·1	0·35	18·6	—	forced	*Marine Rundschau*, 1901, p. 461.
"Holland," "Friesland," "Zeeland," Dutch cruisers, 1896–98	10,000	2 engines, 3 cylinder, triple expansion	(2500 I.H.P. in 2 cylindrical boilers) 7500 I.H.P. in 8 Yarrow boilers		(4,000 in cyl. boilers) + 16,000 in Yarrow boilers = 20,000	(166 + 320) = 486	(31·8) 50	(0·062) 0·46	(20) 23	— ·16 to ·22	— forced	*Schiffbau*, iii., 1901, p. 183; and communicated by the builders.
"Noord-Brabant," Dutch cruiser, 1899	10,000	2 engines, 3 cylinder triple expansion	—	—	25,700	456	56·4	0·39	22	—	forced	*Marine Rundschau*, 1901, p. 461; and communicated by the builders.

PLATE XIVA.

[*To face page* **530.**

The *circulation of water* is produced by the steam and water ascending through the tubes exposed to the greatest heat, while the water descends from the upper to the lower drum through those tubes which are least in contact with the flame. The grate is placed between the two nests of tubes, and the flames find their way up sideways through them.

Consumption of coal per I.H.P. per hour: In larger ships, from 1·8 to 2·3 lb. In torpedo-boats, about 3·15 lb.

I.H.P. per square foot of grate: In larger ships, 17 to 21 I.H.P. In torpedo-boats, 23 to 28 I.H.P.

Ratio of heating surface to grate surface = 45 to 58.

For details of these boilers see Table No. 65.

Recently some Yarrow boilers have been fitted with Howden's forced draught and air heaters. In these the ratio H : R is taken at about = 50.

Casing of the Yarrow Boiler.—Plate XIVA. shows a set of Yarrow boilers for a Chilian warship, with casing and uptake. The arrangement of the boiler fittings can also be clearly seen. On the front of the boiler casing there are light and easily removable dampers, for withdrawing the ashes. The smoke box has double walls, the outer shell being isolated with non-conducting material (asbestos), which is protected by a metal covering.

§ 306. **Normand Boiler** (see Figs. 462 to 466).—This boiler consists of an upper drum, and two lower drums parallel to it, and connected with it by two nests of solid drawn steel tubes. The grate is placed between the two lower drums. The tubes are slightly bent, and can thus expand freely when heated, but the curvature is too slight to permit of the steam collecting in pockets. The two lower drums are connected at one end to the upper by downcast pipes of large diameter, which are outside the casing, and are intended to promote the circulation of water from the upper drum (into which the feed-water first passes) into the lower drums. At the other end the two lower drums are connected by stays or a downcast pipe with branches to each drum. If the stays happen to come within the casing, water circulates through them to keep them cool.

The tubes of the outermost and innermost rows are placed so close together that they form a compact screen or "tube wall." The innermost tubes at the front end of the boiler have a space left between them, from top to bottom. The flames are thus forced to find their way forward from the furnace through these openings, and backwards through the nest of tubes to the back of the boiler, where they strike against a vertical baffle plate, under which they are compelled to pass. At the end of the nest of tubes they pass out either (*a*) forwards, in the same direction, or (*b*) sideways, or (*c*) into the uptake through openings left at the top of

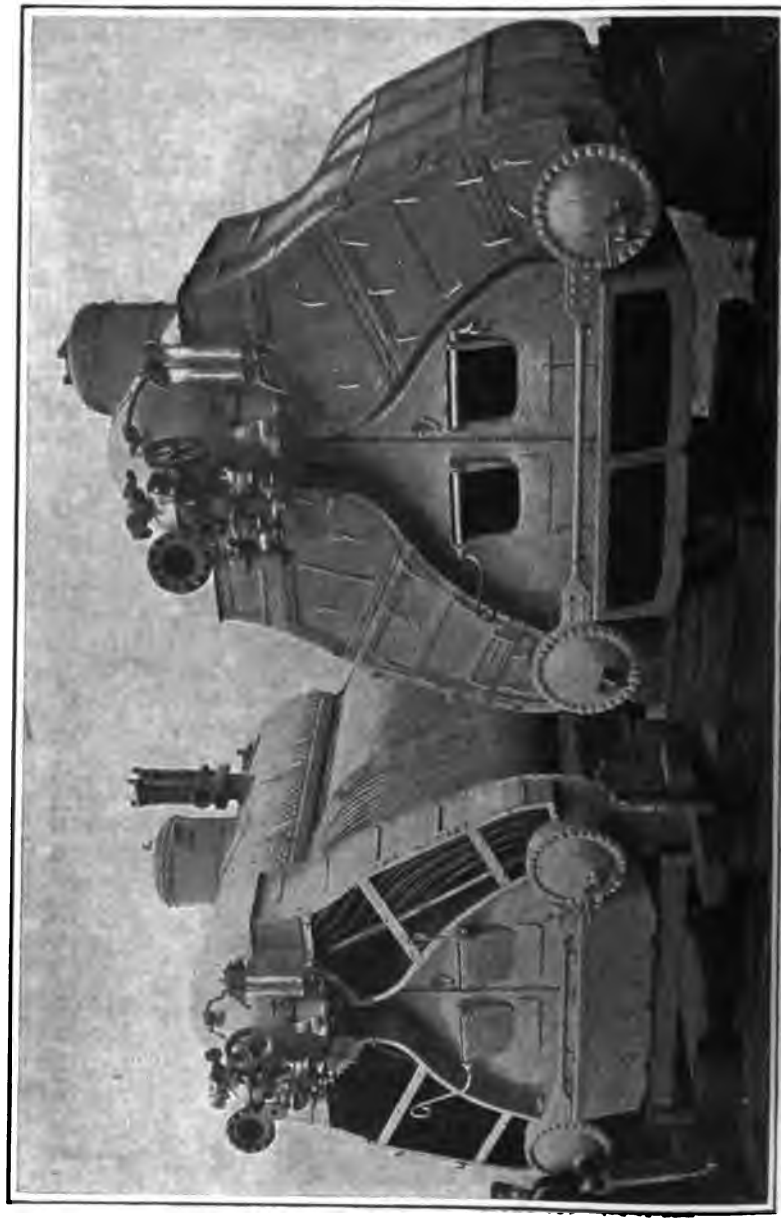


Fig. 462.

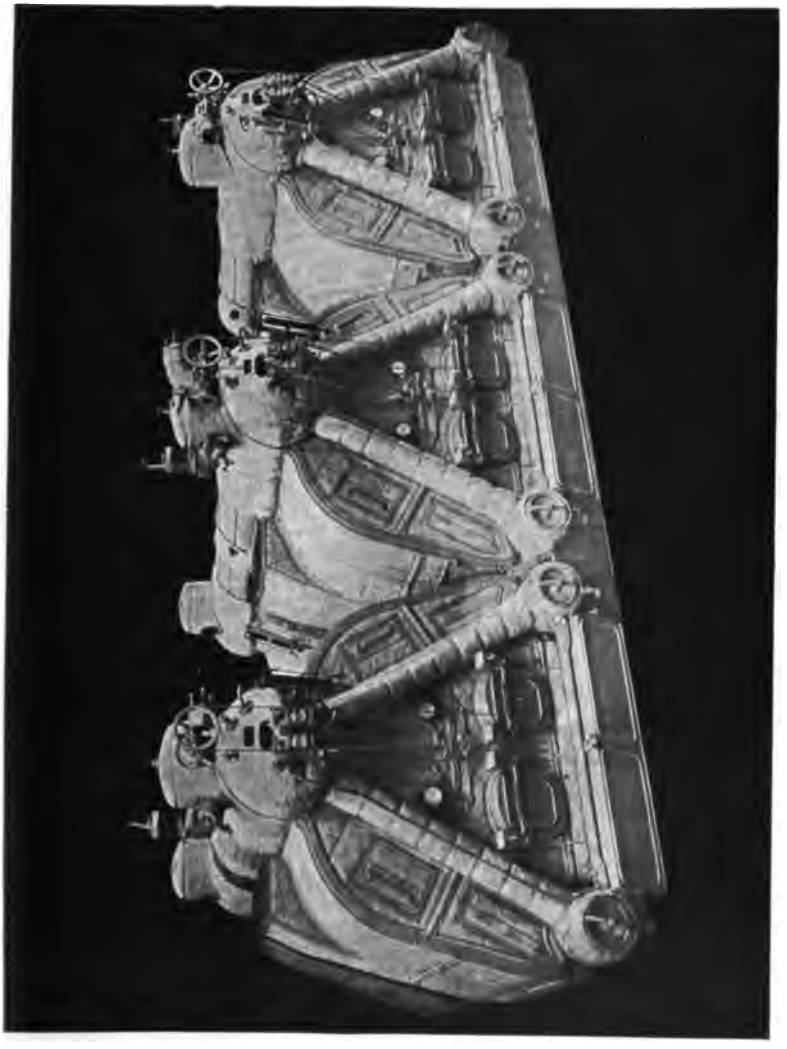


Fig. 463.

the outermost rows of tubes. Various modifications of this arrangement have been tried.

In Fig. 462 the downcast pipes at the further end of the boiler are not seen; but the stays in the front end of the boiler, and the opening in the outer tube wall below the funnel, are clearly shown. Here the flames enter the tubes at the back end of the grate.

Dimensions of a medium-sized boiler: Upper drum, about 3 feet 3 inches in diameter; lower drum, about 1 foot 8 inches; length of upper drum, about 10 feet; number of tubes, 1,000, having $1\frac{5}{8}$ inch external diameter and $\frac{1}{8}$ inch thickness (corresponding to about 3,000 square feet heating surface, and 70 square feet grate area). The mean level of water reaches to about one-quarter of the way up the upper drum. Sometimes the length of the grate is as much as 8 feet 3 inches and more; the amount of space above the grate is so large that, in spite of its considerable length, it is more easily stoked than any other type of boiler.

For details of these boilers see Table No. 66.

Normand Boilers of the Russian Cruiser "Bogatyr" (compare Table No. 66).—This ship is fitted with sixteen Normand boilers, having a total heating surface of 50,700 square feet and 985 square feet grate area. Boiler pressure, 256 lb. per square inch. Of these sixteen boilers, four have each 70 square feet grate area, and twelve have each 58 square feet grate area. Each of the boilers has a heating surface of 3,170 square feet. There are 972 tubes to each boiler, each tube having an external diameter of $1\frac{1}{2}$ inches, and thickness $\frac{1}{8}$ inch. Overall length of each boiler, 11 feet 9 inches. Overall width of the larger boilers, 14 feet 3 inches; of the smaller, 12 feet 8 inches. Total height, including the steam dome, 14 feet 9 inches. Water space per boiler, 140 cubic feet. Steam space, 70 cubic feet. Each boiler has two furnace doors. Length of grate, 7 feet 4 inches. Width of grate in the larger boilers, 9 feet 6 inches; in the smaller, 7 feet 9 inches. Internal diameter of the upper drum, 3 feet 3 inches; thickness, 0·70 inch. Internal diameter of each of the lower drums, 18 inches; thickness, 0·94 inch. Internal diameter of each of the two downcomers, 10 inches. At the back end of the boiler a hollow tie connects the upper to each of the lower drums. A number of dampers are placed in the sides of the ash pit, which admit a plentiful supply of air, and open inwards, so that they are opened automatically by the pressure of air in the stokehold, and close in the same way if any steam escapes from the tubes into the furnace. The steam is taken off through an internal steam pipe opening into the dome, which prevents any priming water being carried off with it.

The flame first enters the nest of tubes near the fire door, passes

TABLE No. 66.—*Normand Boilers.*

Name of Ship.	I.H.P.* Maker's Figures.	Type of Engine.	Number of Boilers.	Steam Pressure lb. per sq. in.		Total Heating Surface H.	Total Grate Area R.	Ratio H : R. $\frac{H.S.}{G.S.}$	I.H.P. per sq. ft. of H.S.	I.H.P. per sq. ft. of G.S.	Consumption of Coal per I.H.P. per hour.	Consumption of Coal per sq. ft. of Grate Area per hour.	Pressure of Air.	Authority.
				At the Engine.	In the Boiler.									
						sq. ft.	sq. ft.				lb.	lb.	in.	
French torpedo boat, No. 185	1,680*	1 engine, triple expansion	1	—	200	1,840	39	47·2	0·91	43·1	1·56	67·4	3½	*Engineering*, ii., 1895, July and Nov.
"Pelorus," third class English cruiser	7,000	2 engines, 3 cylinder triple expansion	8	250	300	15,850	350	45·3	0·44	20	—	—	—	*Marine Rundschau*, 1901, p. 443.
"Bogatyr," Russian cruiser, 1901 "Vulcan" Works, 1901-1902.	20,200	2 engines, 4 cylinder triple expansion	16	240	256	50,700	985·6	51·5	0·40	20	1·43	—	$\frac{3}{8}$	*Marine Rundschau*, 1901, p. 459.

* The figures marked with an asterisk are the results given on trial trips.

through over the tubes in a horizontal direction, and on reaching the end is directed downwards by a baffle plate; finally it passes out, partly through an opening in the tubes, at the upper end of the outermost row, and partly horizontally at the end of the boiler throughout the

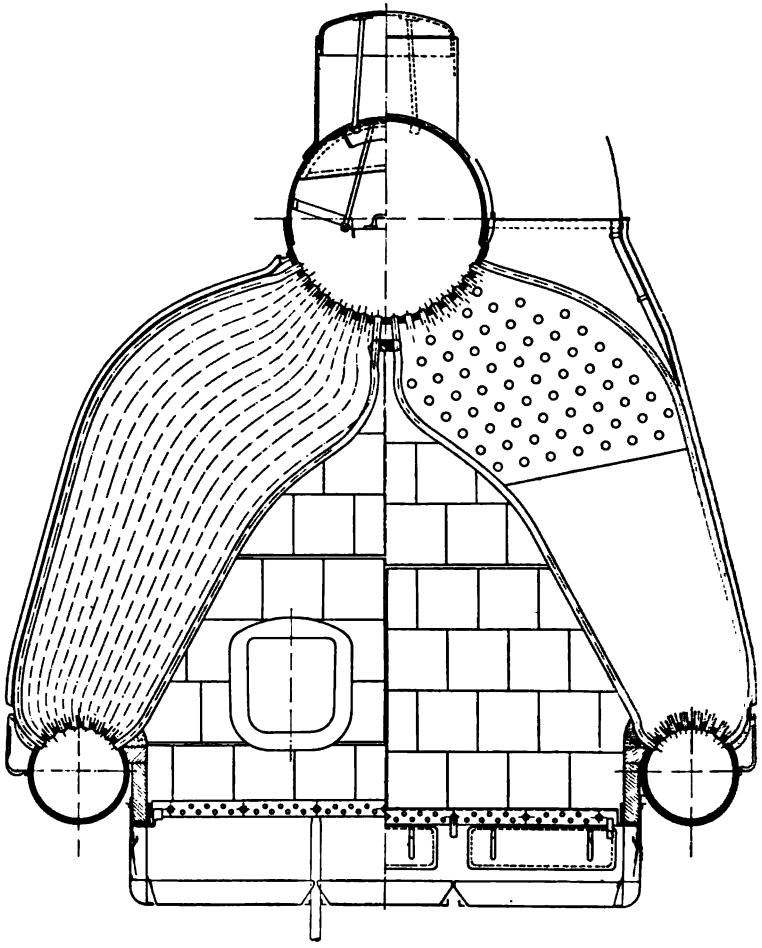


Fig. 464.

whole length of the tubes. The gases of combustion are led through the uptakes into three funnels, the front one being 8 feet 2 inches, and the two other funnels 9 feet 2 inches in diameter. Height of the funnels above the grate, 72 feet.

The construction of the boiler is shown in Figs. 462-466; the con-

struction of the casing and lagging (double metal walls with asbestos cloth laid over them, and secured with iron bands) and arrangement of the boiler mountings is seen in Fig. 463.

Fixing the tubes into the plates of the upper and lower drums is

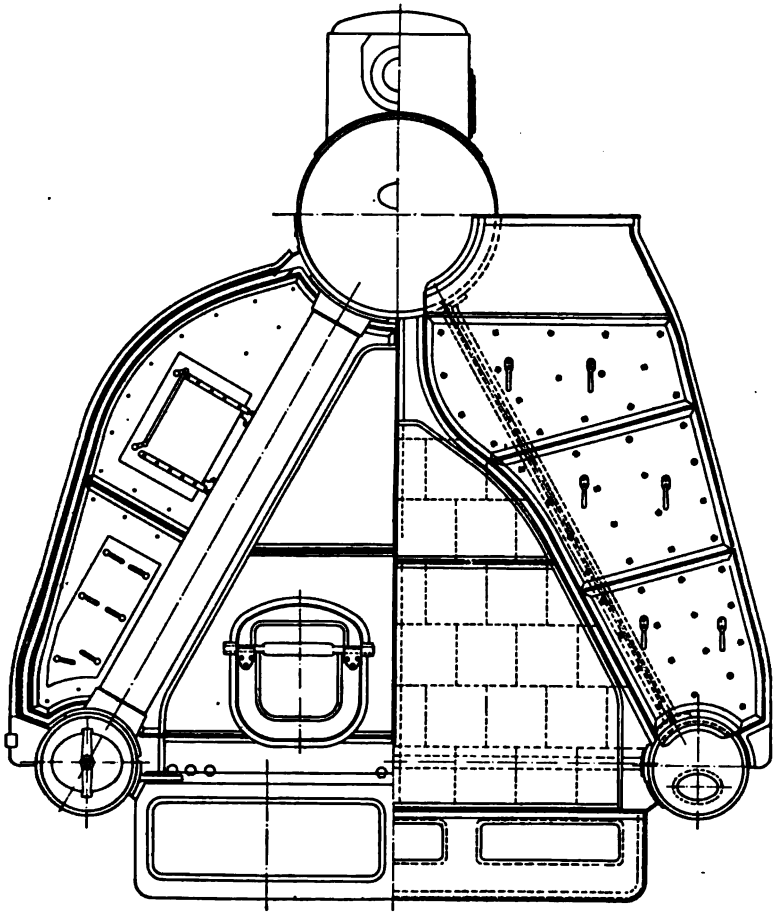


Fig. 465.

effected, as in all boilers with bent tubes, by expanding them into holes, and bell-mouthing the end which projects over. In many cases the holes have spiral grooves, about $\frac{1}{50}$ inch deep. The edge of the hole must be slightly rounded off, where the bell-mouthing takes place, so that there may be no sharp edges.

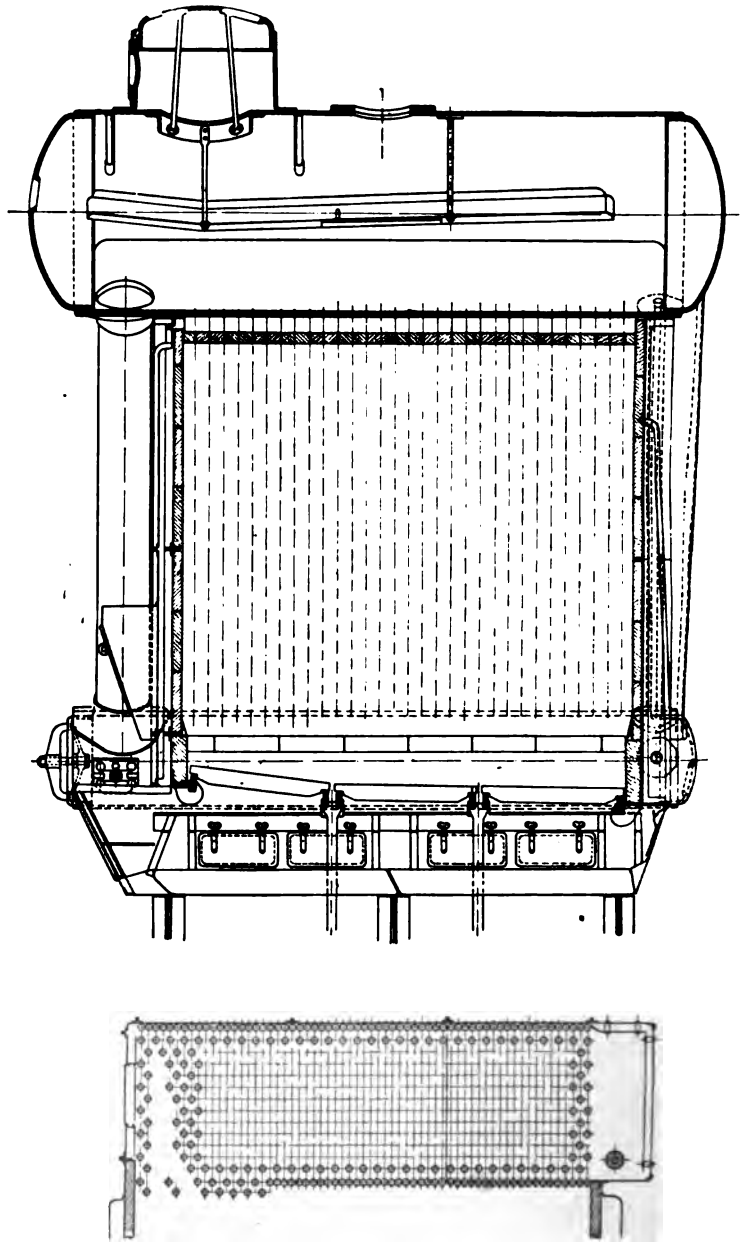


Fig. 466.

The *brickwork* is *secured* to the boiler casing by iron bolts, with T-shaped heads, which are secured at their other end to the casing by nuts or wedges (Fig. 468).

Unrestricted Area of Flues.—The unrestricted opening left to admit the gases of combustion into the nest of tubes is about $\frac{1}{5}$ to $\frac{1}{6}$ of the

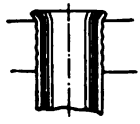


Fig. 467.

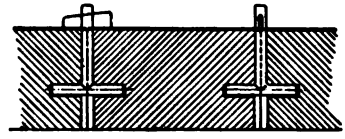


Fig. 468.

grate surface. The unrestricted area for the passage of gases through the nest of tubes, and their exit from the nest, should be not less than $\frac{1}{7}$ and not more than $\frac{1}{5}$ the area of the grate; while the area of the uptake and the funnel is generally about $\frac{1}{5}$ that of the grate.

§ 307. **Small Tube Water-Tube Boilers.**—The various types belonging to this series of water-tube boilers do not markedly differ from each other, and may therefore be treated together.

§ 308. **"Daring" Type Thornycroft Boiler.**—One of the oldest of the better known types is the "Daring" type Thornycroft boiler (see Fig. 469). A grate is situated on either side of the central lower drum. The flames enter the inner nest of tubes at the bottom along the whole length of the grate, and pass out at the top along the whole length of the nest of tubes. On the outer side of each grate the combustion chamber is bounded by a so-called "water wall," *i.e.*, by a double row of tubes, which are bent together so as to form a single continuous wall; these tubes open at their lower end into a lower drum of very small diameter, which is connected through a pipe with the central lower drum. The water, which is fed into the upper drum, passes down into the central lower drum, through a row of tubes in the middle, between the two inner nests of small tubes. Except for these last-named tubes, all the water tubes discharge into the steam space above the water level of the upper drum.

§ 309. **"Speedy" Type Thornycroft Boiler.**—Another older type still is the "Speedy" Thornycroft boiler (see Fig. 470). On each side of the grate is placed a nest of tubes, all the tubes of

which discharge into the steam space of the upper drum. The inner and outer rows of each series form a water wall, except that in order to allow free ingress and egress to the gases into and from the nest of tubes, the inner wall is left open along the whole length of the grate at the bottom, and the outer row along the whole length at the top. The circulation of water from the upper drum—into which the feed-water is fed—to the lower, is by means of downcomers at either end of the boiler. This method of construction has now been abandoned, because the steam collecting above the openings of the small tubes into the upper drum, renders them

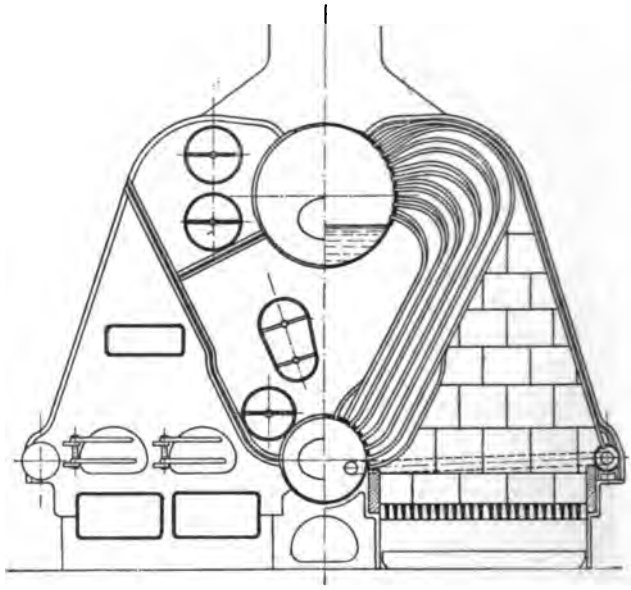


Fig. 469.

liable to overheating, and also when the boiler is laid by, the upper curves of the tubes form "air-pockets," and injury to the boiler is the result.

§ 310. **Thornycroft Boiler.**—For other types of boilers belonging to the systems mentioned in § 307, see Figs. 471, 472, and also Figs. 473, 474. In the *larger sizes*, Figs. 471, 472, these boilers have three lower drums; the middle one is connected to the upper drum by two nests of tubes, and the two outer drums each by a single nest of tubes. At either end of the upper drum a large downcomer pipe leads to each of the two outer lower drums, while the centre one is connected

to the upper drum from end to end by a row of slightly curved down-comer pipes. As none of these pipes are exposed to the heat of the fire, they are able to carry down the water from the upper drum to the lower part of the boiler. Between the lower drums are the two grates. The two innermost and the two outermost rows of each set of tubes are bent so as to form tube-wall rows, with openings at the bottom on the fire side rows, and at the upper ends on the rows nearest the uptake. To make these tubes touch each other, their diameter is increased to

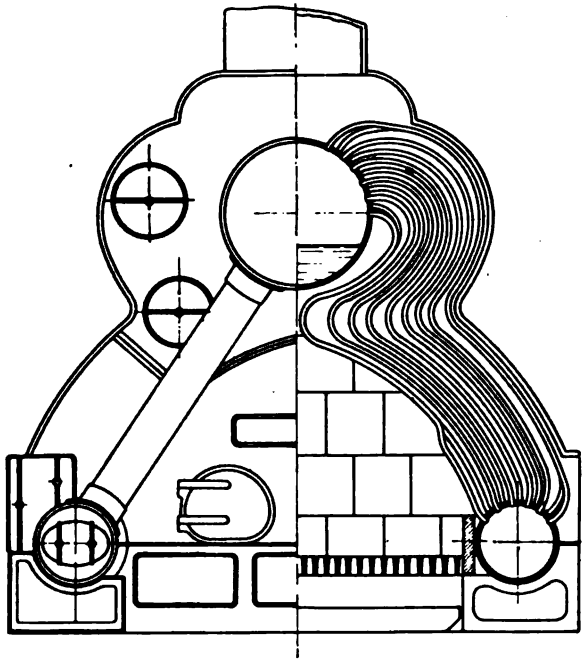


Fig. 470.

about $1\frac{3}{8}$ inch, whereas the diameter of the other tubes is only $1\frac{1}{8}$ inch. The flames, therefore, can only penetrate the nest of tubes at the bottom, where they do not touch each other, and pass out at the top of the "tube wall," where the tubes enter the upper drum. The discharge opening for the hot gases should be as small as possible, and placed immediately below the funnel in the middle of the boiler. The boiler shown at Figs. 471 and 472 is designed to give 1,750 I.H.P.; heating surface of tubes, 3,600 square feet; grate area, 63 square feet; pressure above atmosphere, 230 lb. per square inch.

In the *smaller sizes* (see Figs. 473 and 474) there are only two lower drums, from each of which a nest of tubes leads to the upper drum. The grate is situated between the two nests of tubes. The inner and outer rows of both nests of tubes form, as before, continuous tube walls, which only afford a passage to the hot gases through the inner rows at the bottom, and through the outer rows at the top. The tubes, as

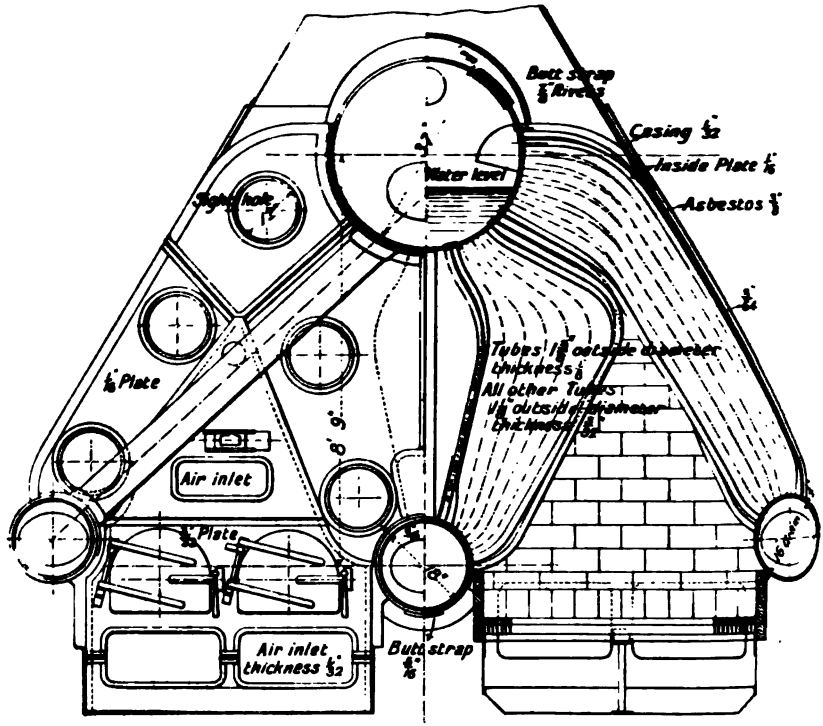


Fig. 471.

before, are solid or seamless drawn steel, galvanised on the outside, and expanded into the upper and lower drums; the tube ends are bell-mouthed on the inside of the drum.

The boiler for a torpedo-boat, shown in Figs. 473, 474, has a heating surface of 1,150 square feet, grate area of 21 square feet, and a working pressure of 145 lb. per square inch. The water level is about one-quarter of the way up the upper drum.

The feed is regulated by a solid drawn W.I. float inside the upper

drum, which rises and falls with the water level, and thus actuates a horizontal rod through a bell crank lever. The rod passes through the front of the boiler, and actuates a slide or lift valve, throttling the admission of feed-water if the water level is high, and admitting it freely if the water level is low. The casing is composed of double plates (see

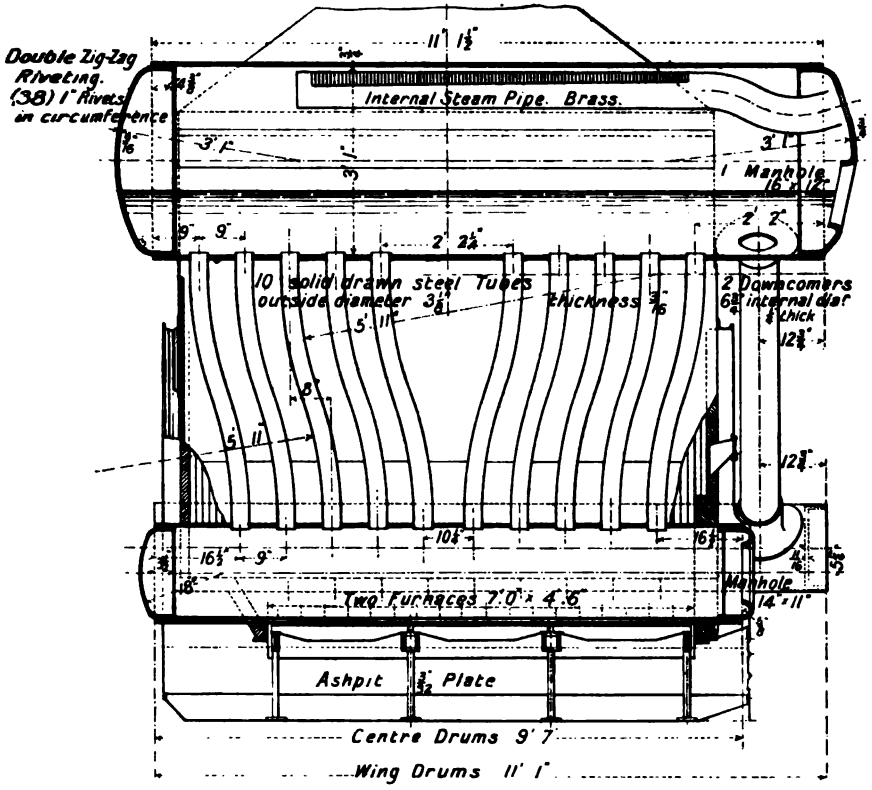

Double Zig-Zag Riveting. (38) 1" Rivets in circumference
Internal Steam Pipe. Brass.
1 Manhole
10 Solid drawn steel Tubes outside diameter 3 1/4" Thickness 3/16
2 Downcomers 6 1/2" internal diaf
Two Furnaces 7'0" x 4'6"
Ashpit 3/4" Plate
Centre Drums 9' 7"
Wing Drums 11' 1"

Fig. 472.

Figs. 473, 474), between which asbestos is packed, the outside being lagged with non-conducting material.

§ 311. **Recent Thornycroft Boilers.**—In recent boilers of this type, in the construction of which Schulz has rendered considerable service, the flue gases, in order to more fully utilise the heat contained in them, are led through specially constructed passages. Such a boiler is shown at Fig. 475. The flames enter the central nest of tubes

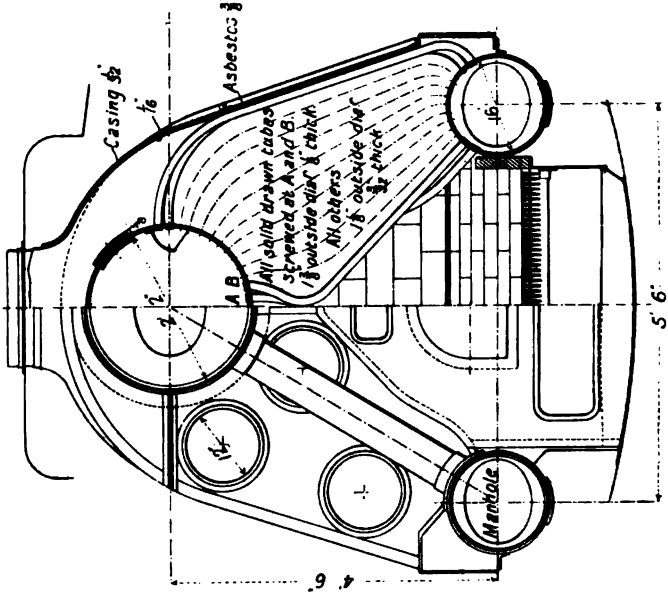


Fig. 473.

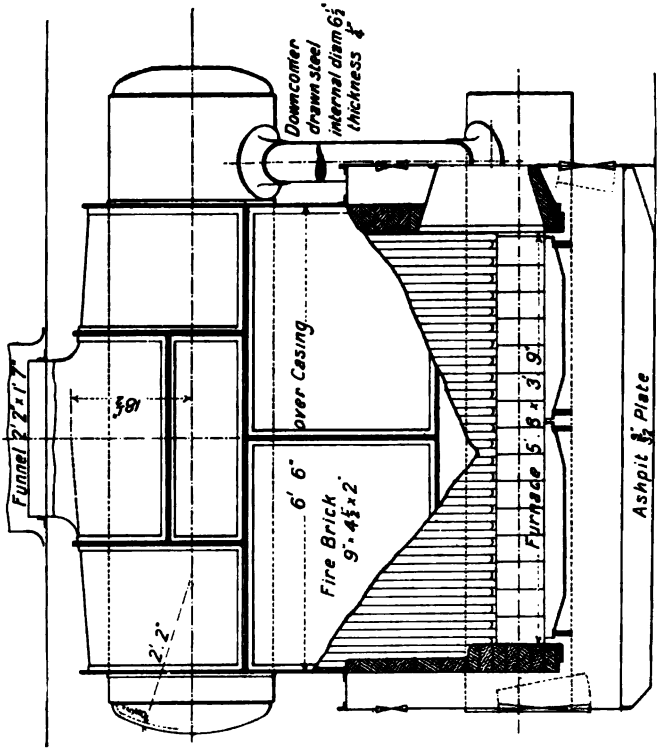


Fig. 474.

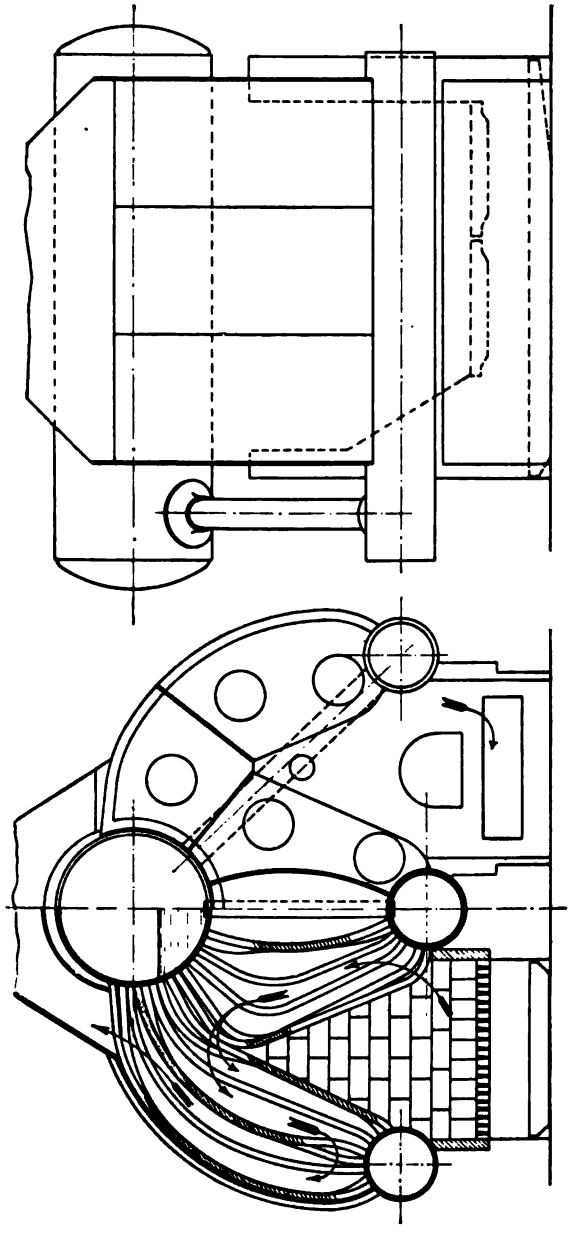


Fig. 475.

TABLE No. 67.—

Name of Ship.	I.H.P.* Maker's Figures.	Type of Engine.	No. of Boilers.	Steam Pressure, pounds per sq. in.		Total Heating Surface in sq. ft.
				At Engine.	In Boiler.	
—	7,000	—	4	—	230	14,400
H.M.S. "Proserpine," third-class cruiser	7,000	2 engines, 3 cylinder triple expansion	8	250	300	19,475
"Missouri" and "Ohio," American battleships, 1900	16,000	2 engines, 4 cylinder triple expansion	12	—	250	53,200
"Novik," Russian cruiser, 1901	17,000	3 engines, 4 cylinder triple expansion	12	—	256	49,300
"Ægir," German coast defence vessel	4,980* 3,680*	2 engines, 3 cylinder triple expansion	— —	— —	— —	16,140 16,140
"Niobe," small German cruiser	8,110* 3,890*	2 engines, triple expansion	— —	— —	— —	21,730 21,730
"Iltis"	1,378* 1,293*	Triple expansion	— —	— —	— —	4,000 4,000
H.M.S. "Handy," torpedo-boat destroyer, 1896	4,600	2 engines, 3 cylinder triple expansion	3	—	185 (max. 210)	9,500
"Eber," gunboat, German Navy; built 1903 at the Vulcan Works, Stettin	1,300	2 engines, 3 cylinder triple expansion	4	about 180 lb. per sq. in.	185	3,900
"M," small cruiser, German Navy; built 1903	10,000	2 engines, 3 cylinder triple expansion	10	200	215	30,000
"Preussen," battleship, German Navy; under construction (1903) at the Vulcan Works, Stettin	16,000	3 engines, 3 cylinder triple expansion	8 water tube 6 cylindrical	185 185	192 192	34,400 15,000
"Mecklenburg," battleship, German Navy; delivered (1903) by Vulcan Works, Stettin	13,600 14,350*	3 engines, 3 cylinder triple expansion	6 water tube 6 cylindrical	185 185	185 185	29,000 15,000

* The figures marked with an asterisk are the results given on trial trips.

Thornycroft, Schulz, and similar Boilers.

Total Grate Area K.	Ratio H : K.	I.H.P. per sq. ft. of H.	I.H.P. per sq. ft. of K.	Consumption of Coal per I.H.P. per hour.	Consumption of Coal per sq. ft. of Grate per hour.	Pressure of Air.	Authority.
sq. ft.				lb.	lb.	inches.	
250	57·2	0·48	27·8	—	—	—	Communicated by the builders.
350	55·7	0·36	20	—	—	3·2	*Engineering*, 1899, ii., pp. 216-218.
960	55·4	0·30	16·6	—	—	—	*Marine Rundschau*, 1901, p. 449.
860	57·5	0·34	19·8	—	—	—	*Ibid.*, p. 459.
273·7	59	0·30	18	—	—	—	*Ibid.*, pp. 556-559.
273·7	59	0·22	13	·156	22·5	—	
415·8	52·3	0·37	20	—	—	—	*Ibid.*
415·8	52·3	0·18	0·93	·173	17·4	—	
81·7	50·1	0·33	16·8	—	—	—	*Ibid.*
81·7	50·1	0·31	15·8	·146	25	—	
188	50·4	0·48	24·3	—	—	3·0	*Engineering*, 1896, i., p. 246.
75	51·4	·33	17·2	—	—	max. allowable 2·56	
580	50·8	·33	17	—	—	max. all. 2·56	
650	52·7	—	—	—	—	max. all. 2·56	
450	33·4	—	—	—	—	max. all. 0·47	
670	—	—	—	·16 at 9610 I.H.P.	—	1·3 †	*Schiffbau*, 1903, No. 2.
300	—	—	—		—	0·35 †	

† When developing 14,350 I.H.P.

at the bottom along its whole length, and turn in an upward direction. After passing the centre of the furnace, they pass in a downward direction under a solid wall of tubes in the centre of the outer nest (extending the whole length of the grate, and from the central upper drum almost to the wing drum in the vertical direction), and thence upwards to the funnel, as indicated by the arrows. A row of downcomers is placed in the middle between the two walls of tubes, inside the central nest. Sectional area of these downcomers about ·03 square inch per square foot of heating surface. The lower wing drums are either provided with special downcomers, or connected to the central lower drum, by horizontal tubes. For particulars of boilers of this type see Table No. 67. The boiler casings are made of thin plate, and stiffened on the front and back of the boiler with angle irons. They are provided with circular openings for cleaning the tubes. Wherever the flames can strike against the casing, it is protected by a fire-brick lining. At a greater distance from the grate asbestos millboard is used. Reckoning from within outwards these parts are covered first with asbestos, then the plating, then an air space, then plating again, and lastly a covering of asbestos cloth, held in place by iron straps. Where the casing is protected by a wall of tubes, the inner asbestos millboard is omitted.

SECTION V.

SMOKE BOX, FUNNEL, AND BOILER LAGGING.

(For arrangements of induced and forced draught see Figs. 481 to 483 and 486 to 489.)

§ 312. **The Smoke Box** is generally made with double walls.

Thickness of inner plate, in small light vessels $\frac{3}{32}$ to $\frac{1}{8}$ inch, in larger vessels $\frac{1}{8}$ to $\frac{1}{4}$ inch, in exceptionally large heavy steamers up to $\frac{3}{8}$ inch. Thickness of outer plate about $\frac{3}{32}$ to $\frac{1}{8}$ inch; in lighter ships not more than $\frac{1}{16}$ inch. Air space between the plates, 2 to 4 inches. The plates are connected to each other by bolts and ferrules.

For *cross-sectional area of smoke box* see page 459.

Doors are fitted in the smoke box for cleaning the tubes; there is generally one for each nest of boiler tubes. To prevent the cold air streaming in through the open fire doors, when cleaning the fires, the uptakes from the different furnaces, or in cylindrical boilers from the different groups of tubes, are frequently divided in the smoke box, and the draught in them controlled by separate dampers. The doors are hung on stout hinges and closed with strong catches; they are not only made double, but have baffle plates on the inside, against which the hot gases impinge on issuing from the tubes. Thickness of the centre plate, from about $\frac{5}{32}$ to $\frac{1}{4}$ inch; of the outer plate, $\frac{3}{32}$ to $\frac{1}{8}$ inch; and of the inner, $\frac{5}{32}$ to $\frac{1}{4}$ inch. The smoke-box doors open either sideways or upwards. In the case of large funnels, iron ladders are provided for the inspection of the uptake and funnel, access being obtained through carefully closed manholes. Compare construction of uptake and funnel for a large warship, at the end of this section, § 316.

§ 313. **Funnel.**—In merchant ships the cross section of the funnel is almost always circular, in war vessels it is often elliptical. (For sectional area see page 459.) The height depends on the construction of the ship. (In the largest fast steamers the height of the upper edge of the funnel above the grate is about 100 feet.) The funnel proper is either connected direct to the uptake, or to the funnel seating and through it to the uptake. The cross section of the funnel seating is

generally square at the bottom, and octagonal or round at the top (see Figs. 476 and 479). Thickness of the funnel plate at the bottom (below the upper deck), $\frac{5}{32}$ to $\frac{1}{4}$ inch; further up (above the deck), $\frac{1}{8}$ to $\frac{3}{16}$ inch; at the top, $\frac{5}{32}$ to $\frac{3}{32}$ inch. From the uptake to the deck coaming the funnel is *invariably* protected by a casing (thickness, $\frac{3}{32}$ to $\frac{1}{8}$ inch; distance between it and the funnel, from 2 to 4 inches), and also above the deck coaming one is usually fitted (of the same thickness;

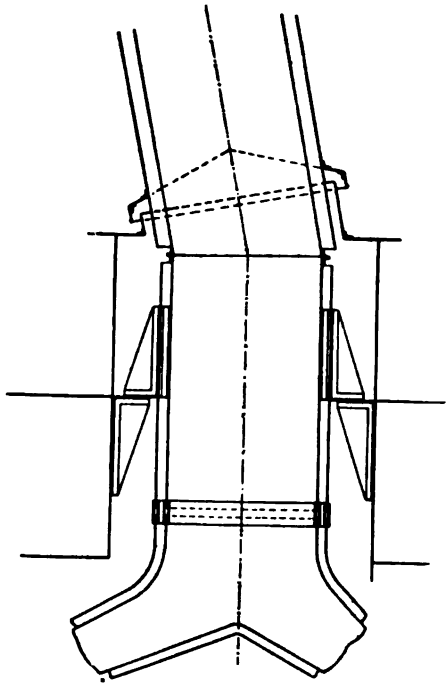


Fig. 476.

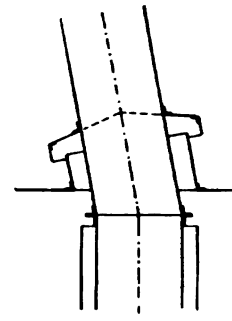


Fig. 477.

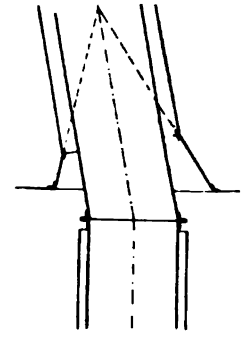
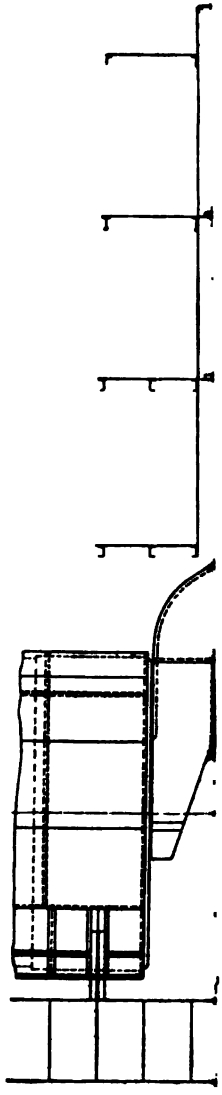


Fig. 478.

distance from the funnel, 4 to 6 inches). The casing is generally smooth-riveted on the outside, with inside butt straps, and the funnel is similarly riveted when there is no casing. Between the casing and the funnel a wide space is often purposely left to make the funnel look bigger. In warships this space is often closed at the top by a metal hood.

§ 314. **Fixing of Funnel.**—The expansion of the metal due to heat must here be allowed for. The uptake is always supported on the boiler, and in large ships the funnel is supported from the funnel



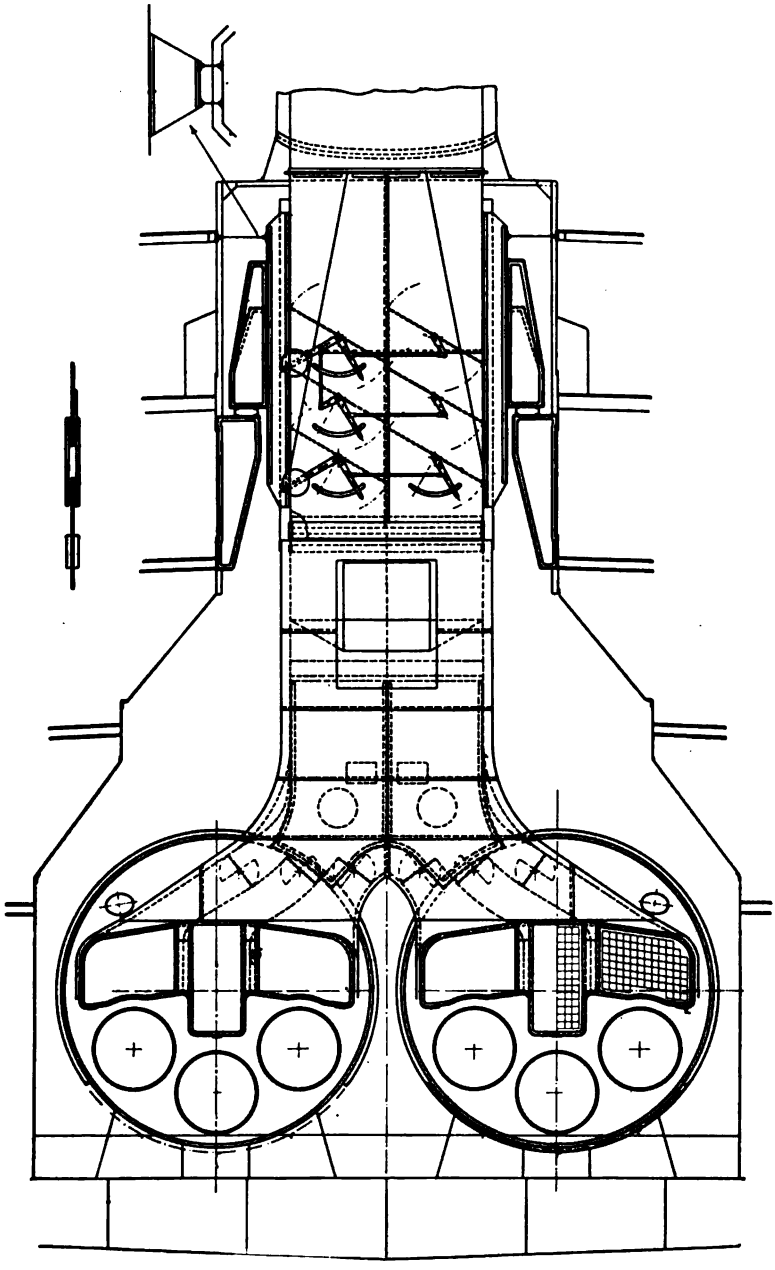


Fig. 480.

casing, being secured above the uptake by means of angle-iron brackets. (In large installations it is supported on angle-iron brackets resting on similar brackets bolted to the funnel casing, see Fig. 476). To allow for the expansion of the uptake between the boiler and the base of the funnel, the uptake below the latter is sometimes cut through, and a double butt-jointed ring is fitted round the part where it is cut, in which the funnel and the casing can expand (Fig. 480). The upper part of the funnel either slides in the casing itself, if the latter is bolted to the deck coaming (Fig. 478), or it expands together with the casing and apron (see below) out of the ring or casing, which is bolted to the deck coaming (Fig. 476). Where the funnel has no casing (Fig. 477), it expands with the apron through the deck coaming.

To prevent water finding its way down the funnel into the boiler-room during bad weather, the hole in the deck casing into which the funnel fits is often protected by a metal apron secured to the funnel or the casing, or occasionally the casing itself may be enlarged at the bottom to fill up the hole completely (Fig. 478). A short distance below the top of the funnel, one, or sometimes two, strong rings are riveted on, to which shackles for the guy ropes are fitted. If there is a casing, the outer ring is firmly bolted to the inner casing by means of bolts and ferrules. The stays, which consist of chains or wire rope, are held with lanyards to the body of the ship, and must be loosened as soon as the funnel gets hot.

§ 315. **Funnel Dampers.**—At the bottom of the funnel a damper is provided for regulating the draught. If there are several boilers discharging into a common funnel, the latter is frequently divided up by longitudinal partitions from top to bottom, so that the smoke from each boiler passes up through a separate channel. In this way the draught is not affected if any one boiler is not working. Each of these divisions has a damper of its own. The smoke from the donkey boiler is also led into the main funnel, but in a separate pipe, and carried up to the top of the main funnel so as to improve the draught when in port, and allow the main funnel to be overhauled. The blow-off pipe from the safety valves is generally carried some way up the funnel, either in front or behind. To facilitate inspection of the funnel, an iron ladder is generally fitted inside it. In small river steamers some arrangement is usually provided for lowering the funnel, corresponding counterweights being provided.

§ 316. **The uptake and funnel for a war vessel** of the large cruiser or battleship type are constructed more or less on the following lines :—

Uptake.—Each boiler has a separate uptake carried up to between decks, and fitted with a separate damper, so that, if desired, any boiler may be isolated. There is a separate damper for each combustion chamber, but these dampers must not impair the free access to the tubes. In the bent parts the thickness of the plate is about $\frac{3}{16}$ inch, elsewhere about $\frac{5}{32}$ inch. Where the uptake passes over those parts of the boilers which are in contact with the steam, an air space of at least $2\frac{1}{4}$ inches is provided. The lower portion of the uptake is bolted to the boiler, and can expand into the lower part of the casing between the armoured decks. The upper part of the uptake is supported between decks by brackets or slings, and can expand between the funnel and the armoured upper-deck. When the uptakes are inclined or at an angle it is sometimes necessary to support or suspend them. The hatchways for the uptake and funnel in the armoured deck are protected with armoured gratings. The deck coamings for the uptake and funnel, which consist of deck beams, or vertical plating, are fitted with brackets to take the armoured gratings. The uptakes are accessible through doors opening inwards.

Funnels.—The thickness of the funnel plates is about $\frac{5}{32}$ inch at the bottom, and $\frac{1}{8}$ inch at the top. They are provided inside with iron stiffening rings. The funnel stays are provided with adjustable shackles, fitted with bronze nuts, to secure the funnel, which is generally attached to the armoured upper deck.

Uptake and Funnel Casings.—Both uptakes and funnels are fitted throughout their length with a casing about $\frac{1}{8}$ inch thick. The casing plates are jointed by means of inside butt straps. The casings are either fitted with removable doors, or are partly bolted together so that the outside of the uptakes may be accessible for cleaning. The spacing between the uptake or funnel and the corresponding casing is generally from 3 to 4 inches. From the deck upwards the spacing between the funnel and casing is generally about 12 inches. The annular passage thus formed is utilised to carry away the hot air, and allows it to pass out below the apron. With water-tube boilers it is best to have dampers below the armoured deck, by means of which, with forced draught, the exit of air from the boiler-room up the annular space round the funnel can be regulated. These dampers are so arranged that they can be worked by hand from some convenient position.

The casings are made air-tight at their lower end, where they join the uptake. To regulate the pressure of air in the boiler-room, and the ventilation of the space between the uptake and casing, slides or dampers are often placed at suitable points in the lower part of the casing. When blowing steam through water-tube boilers, to clean the

external surface of the tubes, these dampers must be carefully closed. The casing of the uptake is lagged with asbestos mats secured in position by outside plating, or by means of iron bands. When the bulkheads round the uptake form at the same time a part of the ventilating shaft, they must be lagged with non-conducting material. The upper part of the funnel casing is often fitted with a ring of round iron about 6 inches from the plating, which is intended to facilitate painting the funnels, putting on the covers, &c. Iron rungs are frequently fitted on the outside of the funnel casing, to form a ladder from the deck to the top of the funnel.

Funnel Covers.—Each funnel is generally provided with a galvanised iron cover made in sections.

§ 317. **Boiler Lagging.**—In *cylindrical boilers* about two-thirds of the upper circumference is covered with a non-conducting material (fossil meal, preparations of cork, &c.) and cased in plating $\frac{1}{16}$ inch thick. The lower portion of the boiler shell is usually covered with an easily removable covering or lagging (such as asbestos fibre, felt, strips of cork, &c.). Felt should only be used if the pressure in the boiler does not exceed 110 to 140 lb. per square inch, otherwise it is liable to get charred. In *water-tube boilers* the boiler casing is made double, the inner plate being from $\frac{3}{32}$ to $\frac{1}{8}$ inch thick (thinner for torpedo-boats); next this is a sheet of asbestos or an air space $\frac{3}{4}$ to 2 inches thick; then another plate from $\frac{3}{64}$ to $\frac{1}{16}$ inch thick; and lastly a layer of non-conducting material or asbestos mats. (Compare water-tube boilers.) Where the flame impinges directly on the boiler casing, the inner plate is further protected with asbestos millboard.

SECTION VI.

FORCED DRAUGHT.

§ 318. **General Remarks.**—Forced draught may be divided into—

1. Draught produced by sucking out the gases of combustion, or "induced draught."
2. Draught produced by creating an air pressure in the ash-pit, or "the closed ash-pit system," as in Howden's system.
3. Draught produced by creating an air pressure in the stoke-hold, or "the closed stokehold system."

§ 319. **Induced Draught** (Ellis' and Eaves' system, see Figs. 481 to 483).—This system is generally so arranged that natural draught can also be used. The gases of combustion are sucked out of the smoke box through a special tube SS, and then forced through the tube D into the funnel by means of a fan. A vacuum is thus formed below the fan, and the air for combustion is caused to flow rapidly through the fire. Before being led into the ash-pit, the air is heated in an air-heater placed in the smoke box. This apparatus, which is now usually arranged vertically, consists of a series of parallel pipes (about $2\frac{3}{8}$ inches internal diameter) through which the hot gases from the combustion chamber pass on their way to the smoke box. The air enters the heater V at the top, through dampers KK, circulates round the pipes, and passes out through passages W to the fire F. The main body of the air enters the grate (see Fig. 484) through the horizontal dampers, and some through the damper and slots above the grate. The ash-pit doors are perforated with holes, so that a little cold air may be admitted, to keep down the temperature of the grate. To prevent too much cold air reaching the smoke box while cleaning the fires, it is often partitioned off, and the gases from each fire ascend through a separate passage, which can be separately controlled by a damper. (In Fig. 481 these dampers, which lie immediately above the air-heater, are not shown.) In double-ended boilers the common combustion chamber is divided in two by a firebrick wall.

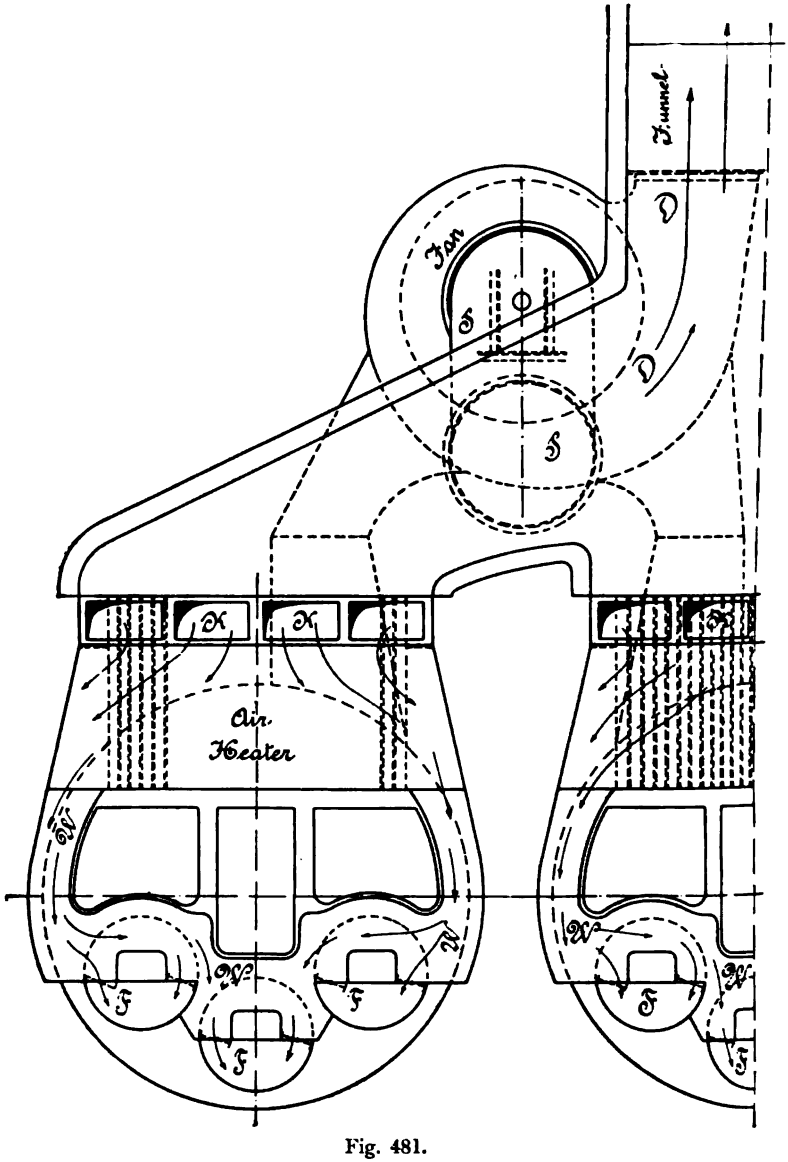


Fig. 481.

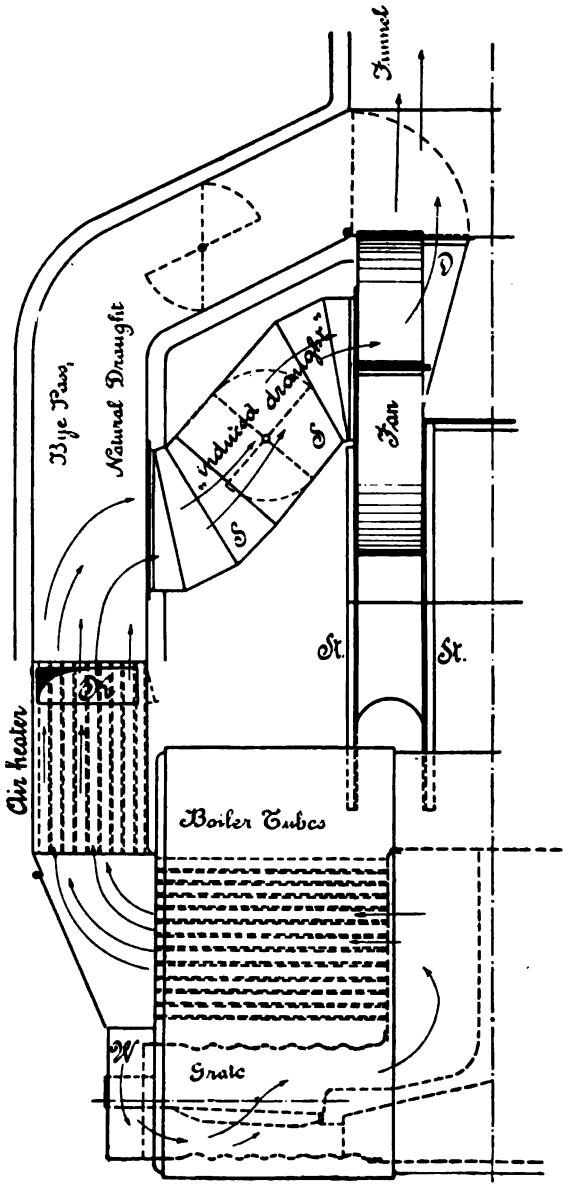

Funnel
D
Induced draught
Fan
S
S
St.
St.
Bye Pass
Natural Draught
Air heater
Boiler Tubes
W
Grate

Fig. 482.

"Serve" tubes (see page 516), which allow of a considerably smaller ratio of heating to grate surface, are sometimes used; if the tubes are of the ordinary kind, "retarders," *i.e.*, spiral coils composed of metal strips, are often fitted into the tubes. The construction of the boilers is the same for induced as for natural draught.

For plain tubes with retarders, the ratio of

$$\frac{\text{heating surface}}{\text{grate surface}} = \text{from } 40 \text{ to } 45.$$

For "Serve" tubes, the ratio of

$$\frac{\text{heating surface}}{\text{grate surface}} = \text{from } 28 \text{ to } 35.$$

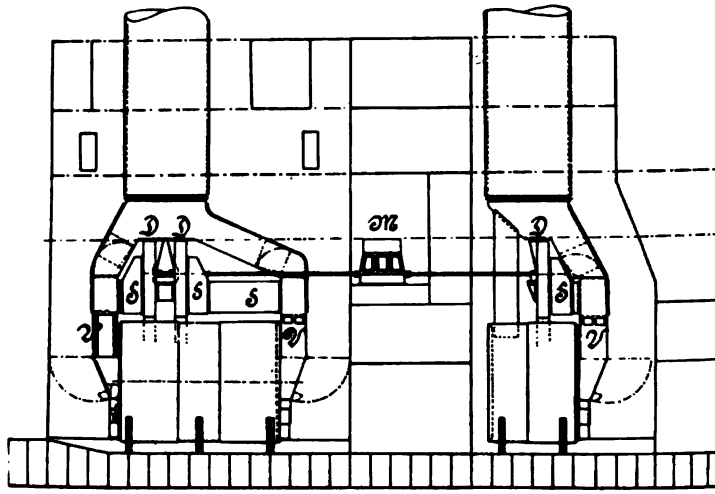


Fig. 483.

Serve tubes may thus be made much shorter than "the ordinary" plain tubes, but they have the disadvantage of being very difficult to clean, and for this reason they have not been fitted so frequently of late years.

Combustion per square foot of grate area per hour (with moderate rates of forcing)—

Fast steamers and mail steamers, 23½ to 25½ lb.
Cargo and passenger boats, 20½ lb.
Cargo steamers, 17½ to 18½ lb.

These are the figures recommended by the patentees, but they are generally exceeded in actual practice.

Temperature of the gases in the fan, 430° to 530° Fahr.
Temperature of the air after passing through the heater, about 300° Fahr.
Vacuum in the fan (with high rates of forcing), from 2 to $2\frac{3}{4}$ inches of water.
Vacuum above the fires, from 0·5 to 0·8 inch of water.
Vacuum in the ash-pit, from 0·25 to 0·4 inch of water.

Diameter of the fan up to 8 feet 6 inches; maximum number of revolutions, with high rates of forcing, from 250 for very large fans, up to 400 per minute for smaller fans. The fan casings are supported by the boilers, and connected to the uptake (for the supports see *s*, Fig. 482); the engines M for driving them are situated between decks (Fig. 483). As the fans tend to rise with the casing as it becomes heated, a universal joint must be provided in the shafting. Compound or two-cylinder engines are used, and one often suffices to drive several fans. The engines are generally completely enclosed to protect them from dust, and to keep them efficiently lubricated. Air is usually admitted on one side of the fan only: the fan centre is generally made of cast steel, the thickness of the side plates is from $\frac{5}{32}$ to $\frac{7}{32}$ inch, and that of the vanes from $\frac{3}{32}$ to $\frac{1}{8}$ inch. Number of vanes, 10 to 16, but half of them are frequently short, and fixed to the periphery of the fan. Fig. 485 shows a fan 7 feet in diameter, driven by a compound single-acting engine. Diameter of H.P. cylinder, $6\frac{1}{2}$ inches; of L.P. cylinder, $11\frac{1}{2}$ inches; stroke, 7 inches. For further details of these engines see § 275 *et seq.*

§ 320. **Howden's System of Forced Draught** (compare § 274). —I.H.P. generated per square foot of grate surface: for fast steamers, working for short periods, 16 to 17; for longer voyages, with inferior kinds of coal, 14 to 15. Ratio of heating surface to grate surface, from 38 to 42 to 1. The higher the rate of forcing, the shorter the life of the plant. For arrangement see Figs. 486 to 489. The fan V is driven by the steam engine M, attached to the forward bulkhead of the engine-room. The air is led into the stokehold through a passage K, in which a damper is fixed at the place where it passes through the engine-room bulkhead. The passage opens into a cross channel or collector Q, which serves to supply the after furnaces with air; from this another passage K_1 passes to the front, and also opens into another cross channel Q supplying the forward furnaces. From these passages the air passes through the dampers E into the air-heater R, and circulates

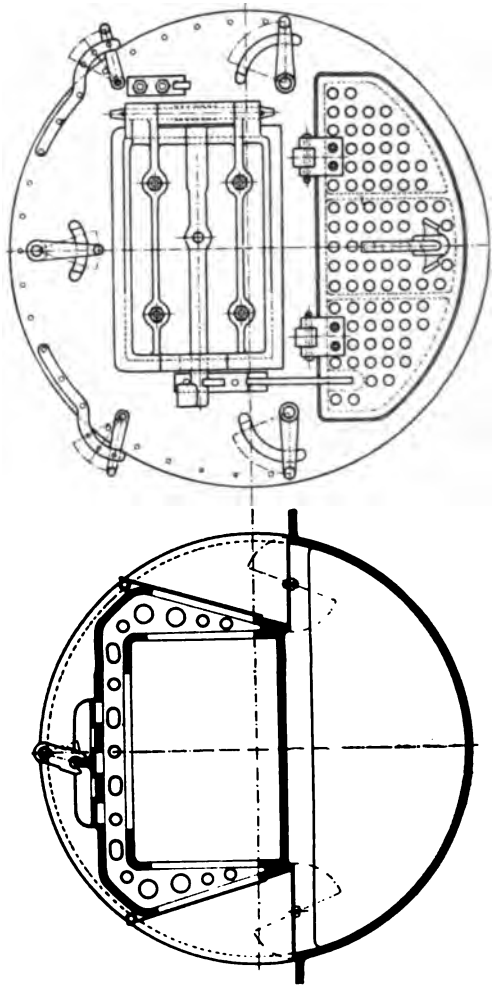


Fig. 484.

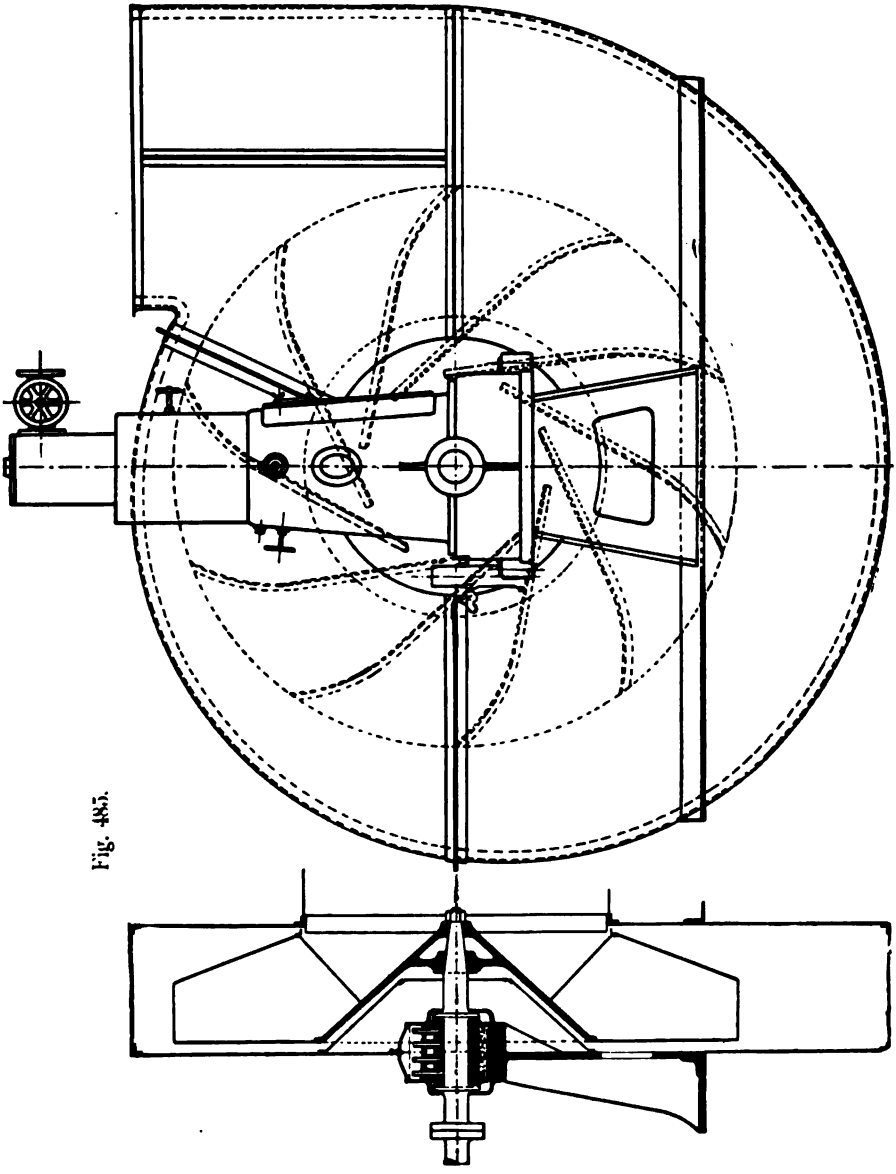


Fig. 485.

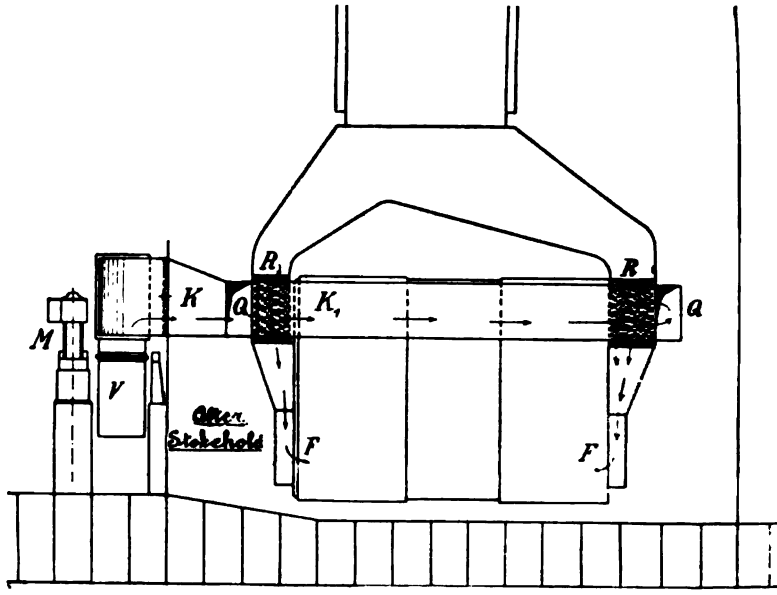


Fig. 486.

Looking on to the Forward Bulkhead of Engine-room.

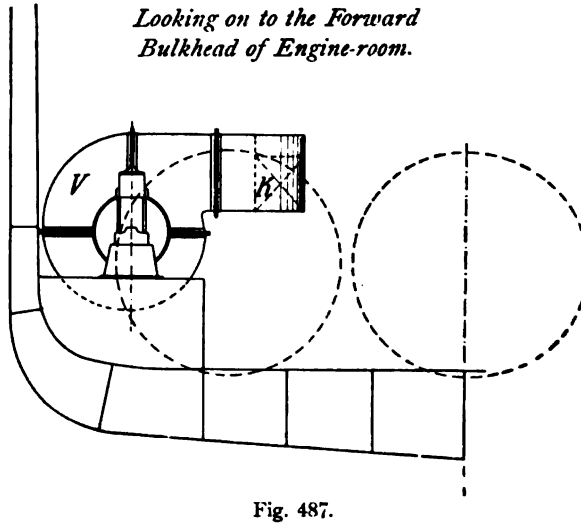


Fig. 487.

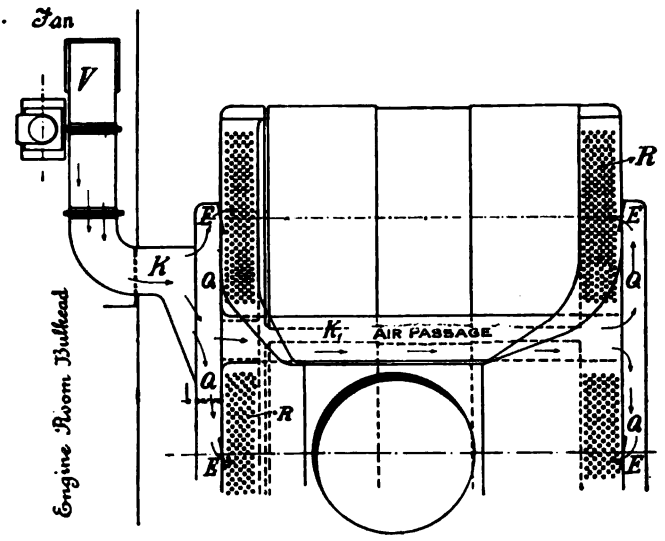


Fig. 488.

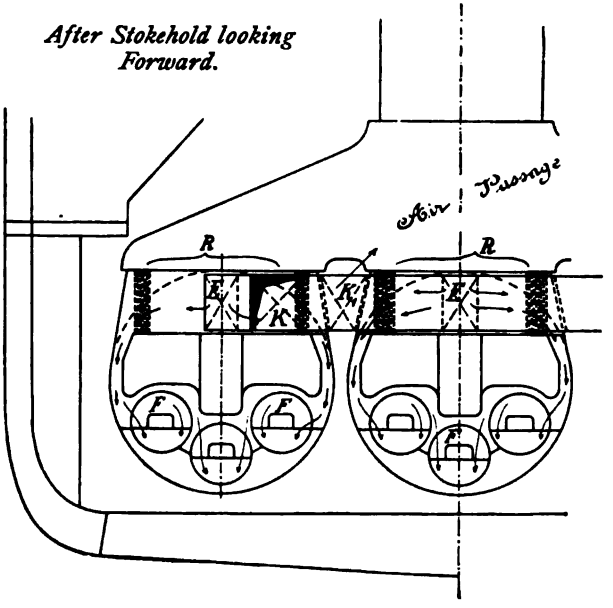


Fig. 489.

round the vertical tubes composing the heater, which are heated by the hot gases. After leaving the heater, the heated air is led through passages at the side of the smoke box to the air passage surrounding the furnaces, and enters the latter through adjustable dampers (in the same way as shown in Fig. 484 for induced draught). These dampers are so arranged that they close automatically when the fire doors are opened, in order to prevent the flames from entering the stokehold, and injuring the firemen.

Sometimes the fans are fitted above the boilers, and mounted on girders riveted to the ship; the engines are then placed between decks; a universal joint is here unnecessary, as both engines and fans are secured to the ship. Diameter of the fans, up to 8 feet 6 inches. Number of revolutions, about 250 per minute. Pressure of air in the air passages, about 2 inches of water. Temperature of the hot air, from about 212° to 300° Fahr. The engines are double-acting, single-cylinder or compound.

§ 321. **Closed Stokehold System.**—In warships with cylindrical, locomotive, or water-tube boilers, forced draught is almost always* obtained by creating a pressure of air in the stokehold. The latter is made as air-tight as possible, and, for purposes of ingress and egress, fitted with an air-lock, *i.e.*, double doors, one of which must not be opened until the other has been closed. The fans are sometimes placed above the armoured deck, and force the air through passages, or direct from the casing, into the stokehold. They may be fitted in the stokehold itself, or hung from the deck, in which case the air is delivered from the fan direct into the stokehold. In both cases the fans draw the air from ventilators fitted on the deck. The speed of the air in the suction trunk about 20 feet, in the delivery pipe about 30 feet per second. A damper must be fitted in the suction or delivery pipe of each fan, to prevent the escape of air through any fan which is not working. The air must be equally distributed throughout the stokehold, which may be done, if necessary, by special baffle plates, and the *personnel* of the stokehold must not be inconvenienced by the fans.

For the necessary quantity of air, shapes of the vanes, power of fan engines, &c., see § 275 *et seq.* External diameter of fan wheel, 4 to 6 feet. Number of revolutions, up to 600 per minute. The air may be sucked into the fan on one or on both sides. Diameter of the intake, about 0·7 × external diameter of fan wheel with single suction, and about 0·5 × external diameter of fan wheel with suction on both sides. Width of fan at periphery, 0·13 to 0·16 × external diameter of fan wheel. Thick-

* The closed ash-pit system of forced draught, or what is known as "under air blast," is now seldom used, apart from Howden's system, on mercantile steamers.

ness of fan casing plates, $\frac{3}{64}$ to $\frac{1}{16}$ inch; of the vanes, about $\frac{1}{16}$ inch. The boss may be of bronze or cast steel.

Pressure of Air.—Even at the highest rates of forcing this is never above 4 inches (and this *only* with locomotive and water-tube boilers). As a rule it does not exceed $2\frac{1}{2}$ inches for water-tube and locomotive boilers, and 1 inch for cylindrical boilers.

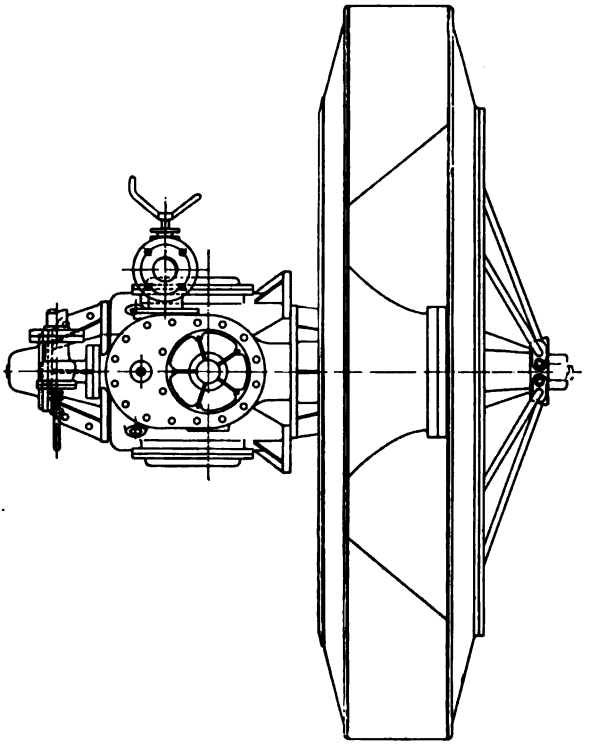


Fig. 490.

Fan Engines.—These may be single or double acting, one-cylinder, two-cylinder, or compound engines. "Chandler" engines are largely used; they are single-acting, wholly enclosed, so that the cranks work in an oil bath, and have long bearings lined with white metal. The exhaust steam is led into the auxiliary exhaust pipe, and the pipe carrying steam to the fan engines must be fitted with a sufficient number of drain-cocks. For data of actual vessels see Tables Nos. 68 and 68A.

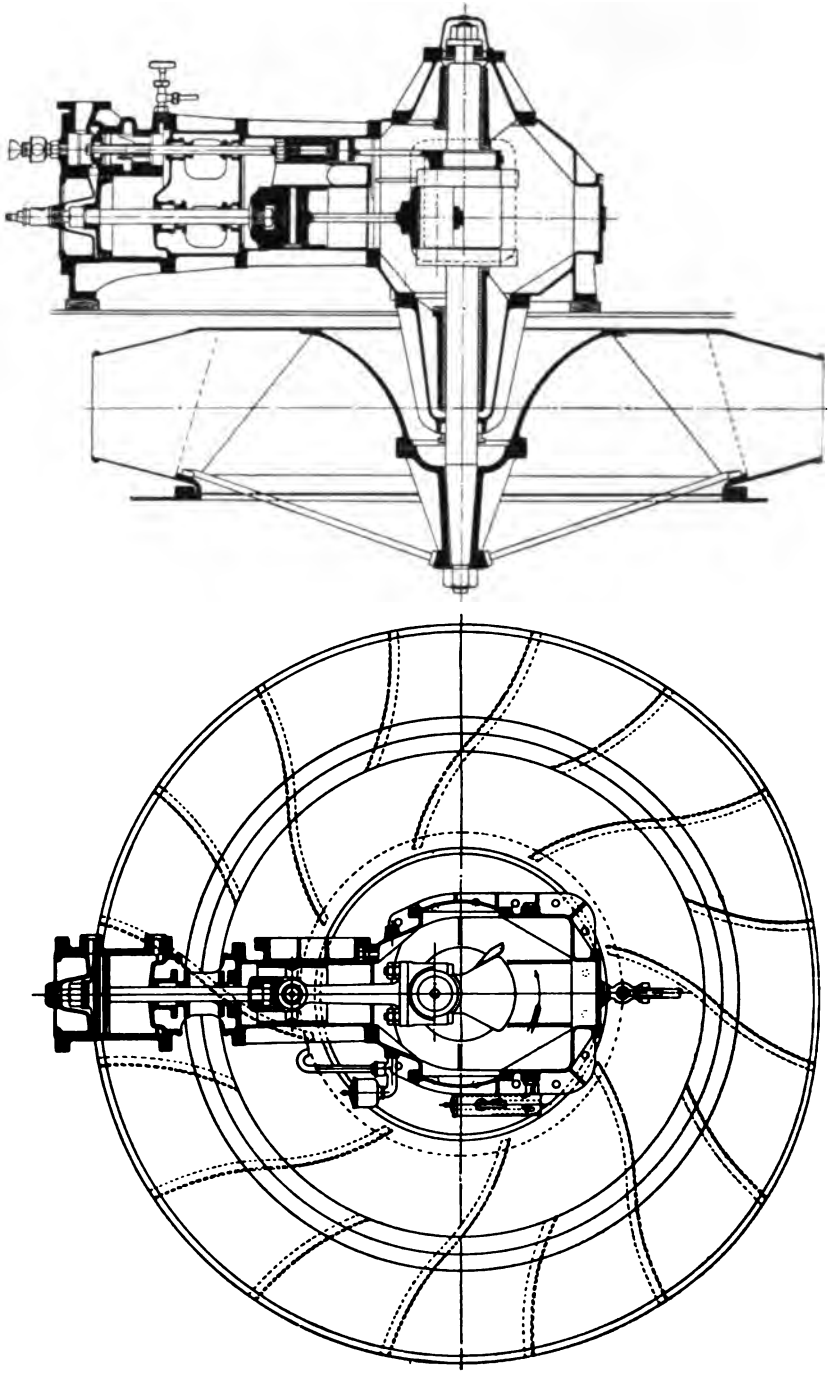


Fig. 401.

TABLE No. 68.—*Examples of the Closed Stokehold System.*

Name of Ship.	—	"Brooklyn."	"Minneapolis."	"Alabama."	"Handy."	"Bogatyr."
Type of ship	Turkish torpedo-boat	American armoured cruiser	American protected cruiser	American ironclad	Destroyer	Protected cruiser
I.H.P. of main engines	1,300	18,248	20,366	11,227	4,500	20,200
Type of boiler	locomotive	cylindrical	cylindrical	cylindrical	Thornycroft	Normand
Number of boilers	1	5 double, 2 single-ended	8 double-ended	8 single-ended	3	16
Working pressure above atmosphere in boiler—pounds per square inch	200	175	163 at engine	190	225	256
Total heating surface H—square feet	1,560	33,400	48,200	21,680	9,485	50,720
Total grate area R—"	35·3	1,000	1,456·7	697	188·8	985
Height of funnel above grate—feet	16·4	100	70	86	—	72
Number of fans	1	12	16	8	3	22
Diameter of fan wheel—inches	63	60	60	60	66	67
Width of fan wheel—"	4½	18	—	14	—	9 (external)
Fan engines: Diameter of cylinder—"	4·7	2 of 5 in. diam.	—	2 of 5 in. diam.	—	7¼ "
Fan engines: Stroke—"	4½	4	—	4	—	4¾ "
Fan engines: Revolutions per minute	—	590	584	480	—	—
Fan engines: Power—I.H.P.	—	22	16·2	13·5	—	—
Total power of all the fan engines—"	—	264	260	108	—	—
Air pressure in stokehold—inches water	—	2½	1	1 1/16	3	⅜
I.H.P. per square feet grate area	36·6	18	14	16	23·7	20·4
Authority	*Zeitschrift des Vereines deutscher Ingenieure*, 1889, p. 1139.	*Journal of the American Society of Naval Engineers*, vol. viii., No. 4.	*Journal of the American Society of Naval Engineers*, vol. vi., No. 4.	*Schiffbau*, ii., April 23, 1901.	*Engineering*, 1896, i., p. 246.	*Engineering*, 1903.

TABLE No. 68A.

Boilers with closed Stokehold for the Imperial German Navy.

Type of Ship.	Gunboat, in course of Construction, 1903.	Small Cruiser, in course of Construction, 1903.			Large Cruiser, in course of Construction, 1903.	Battleship, in course of Construction, 1903.	
Main engine - - I.H.P.	2 × 650	2 × 5,000			3 × 6,333	3 × 5,333	
Type of boiler - - - -	Schulz	Schulz			Original Dürr	Schulz	Cylindrical
Number of boilers - - -	4 single-ended	10 single-ended			16	8	6
Working pressure in boiler, lb. per square inch - - - - -	185	213			206	192	
Total heating surface H sq. ft.	3800	30,000			52,700	34,000	15,000
Total grate surface R sq. ft.	75	580			1250	650	450
Number of fans - - - -	2	2	2	1	16	12	
Diameter of fan wheel inches	43	78	70	65	63	63	
Fan engine { Diam. of cylinders inches	H.P. 2·36/L.P. 3·93	6·8/10·2	5·9/8·8	5·5/8·2	H.P. 5·1/L.P. 8·2	H.P. 4·9/L.P. 8·6	
Fan engine { Stroke - - - ,,	3·5	5·9	5·9	5·5	4·7	4·3	
Maximum pressure of air in stokehold* - - - - inches	2½	2½			2	2½	½
I.H.P. per sq. ft. grate surface -	17·2	17			15·2	—	

* The figures given are the maximum pressures allowable.

SECTION VII.

BOILER FITTINGS AND MOUNTINGS.

§ 322. **Boiler Safety Valves.**—The dimensions of these valves must be so calculated that during continuous firing with the stop valve closed, all the steam generated can escape through them without the pressure rising more than 10 °/₀ above the working pressure. The *sectional area* of the safety valves thus depends upon the amount of heating surface, the boiler pressure, and the method of firing (whether with natural or forced draught), and also upon the construction of the valves themselves. Safety valves must be so arranged that they lift quickly when the maximum allowable working pressure has been exceeded, and close rapidly as soon as the pressure has dropped again to the working pressure. In boilers worked with natural draught, experience has shown that the total internal area of the safety valves is large enough, if calculated from the data given in the following table: *—

TABLE No. 69.

Sectional Area of Boiler Safety Valves.

Pressure above Atmosphere, pounds per square inch.	70	85	100	115	130	140	160	170	185	200	215	230
Unrestricted sectional area in sq. inches per 100 sq. feet of heating surface	1·89	1·62	1·46	1·24	1·14	1·04	·951	·865	·808	·78	·75	·736

* See C. Hartmann, "Official Regulations respecting the Registration and Testing of Marine Boilers."

If the boiler is worked with forced draught, or the grate area is more than $\frac{1}{3\,5}$ that of the heating surface, the total sectional area of the safety valves must be correspondingly increased. In ships, the boilers of which have to be tested according to American regulations, the total area of the safety valves $= \frac{1}{4\,5\,2}$ of the total grate surface. Marine boilers must, in accordance with regulations, always have at least two safety valves. The internal diameter of the valves does not as a rule exceed 4 inches. If the total sectional area required for the safety valves of one boiler exceeds about 25 square inches, it is divided between three, four, or more valves of equal size.

§ 323. **Load on Valves.**—The *load* on each valve is now almost always produced by direct-acting steel springs (see Figs. 492 to 495) of either square or round section. The dimensions of the springs must be so selected that with the normal load on the valve they are compressed to the extent of at least $\frac{d}{4}$ to $\frac{d}{2}$ (see below), the maximum shearing stress upon the material must not meanwhile exceed about 12·5 tons per square inch.

$$d = \sqrt[3]{\frac{S \times D}{C}}$$

Here d = thickness of steel spring of square or round section in inches.

D = mean diameter of coil in inches (centre to centre of coil).

S = load on the spring in pounds.

C = constant, = 8,000 for round steel, 11,000 for square steel, or for naval work 11,000 and 15,000 respectively.

And also if—

f = compression of the spring by the load S in inches.

n = number of complete coils.

t = thickness of steel spring of square or round section in inches.

a = constant = 1,400,000 for round steel, 2,000,000 for square steel.

then $f = \frac{D^3 \times S \times n}{t^4 \times a}$

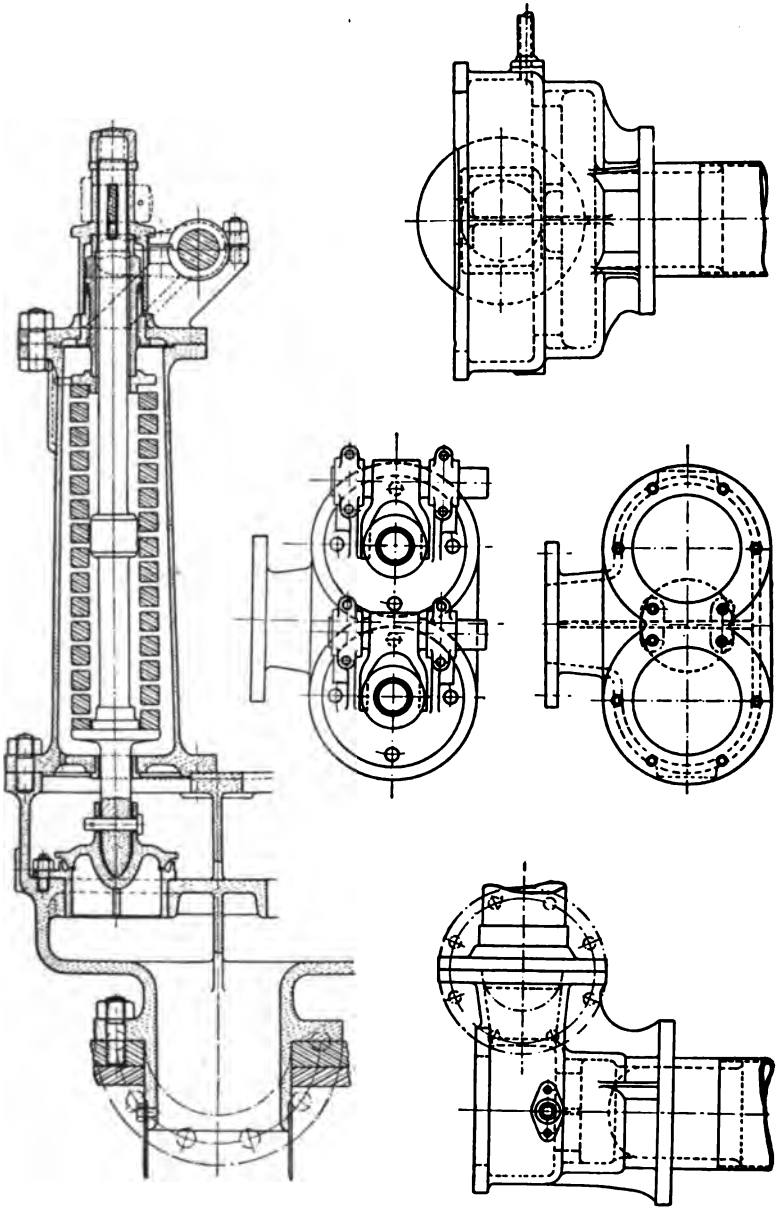


Fig. 492.

The load s on the spring, that is, on the valve, should be

$$s = (d + \cdot 08)^2 \frac{\pi}{4} \cdot p$$

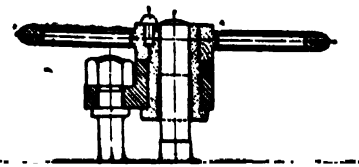
Where d = internal diameter of the valve in inches.
·08 = width of valve seat (inches).
p = maximum allowable steam pressure above atmosphere in pounds per square inch.

Only those coils of the spring must be taken into account which are free and do not touch each other, under normal conditions of loading. When the spring is loaded, there should be a space of from $\frac{1}{8}$ to $\frac{3}{16}$ inch between each of the coils.

§ 324. **Safety Valve Casings.**—The safety valve casing may be made of cast iron, bronze, or cast steel. Figs. 492 to 495 show bronze safety valves as used on warships. The valve seats are always put in separately, if the casing is of cast iron or cast steel, and secured with three or four studs. The valve has three or four guiding ribs or feathers. The seat is generally flat, and not more than ·10 inch wide. The spindle and seat are made as shown in the drawing, so that the escaping steam may strike against the overhanging rim of the valve, and thus hold the latter wide open. The bottom end of the valve spindle should always be situated somewhat below the actual level of the valve seat. To prevent any jamming of the valve spindle, it should be allowed ample "play," not only at the bottom of the spring casing, but also within the spring and the socket at the top.

The cap fitted over the spindle is connected to it by a key or cotter in such a way that the spindle can move freely, should the cap become jammed.* It also prevents any interference with the spindle from the outside. The cotters should be secured by a padlock, or sealed with wire and lead by the surveyor. In order to twist the valve easily, the socket should be fitted with a hexagonal nut, or with two lugs opposite each other. The ***easing gear***, which acts upon the cap by means of a small lever, is arranged so that it can be worked from either the engine-room or stokehold. The safety valve blow-off pipe is carried up alongside the funnel, and has a sectional area equal to, or rather smaller than (about 0·66) the total area of the valves opening into it. The condensed water which collects in the casing above the valve must be carried off by a $\frac{5}{8}$ to 1 inch drain pipe into the bilge, or to a tank. This pipe must never be shut off. In warships the safety valve is often united in the same casing with the

* In Fig. 492 the clear space under the cotter has been omitted in error.



main or auxiliary stop valve, so that one hole only in the boiler is necessary for both (see Figs. 493 to 495). This is especially desirable in water-tube boilers, where space in the upper drum is much restricted.

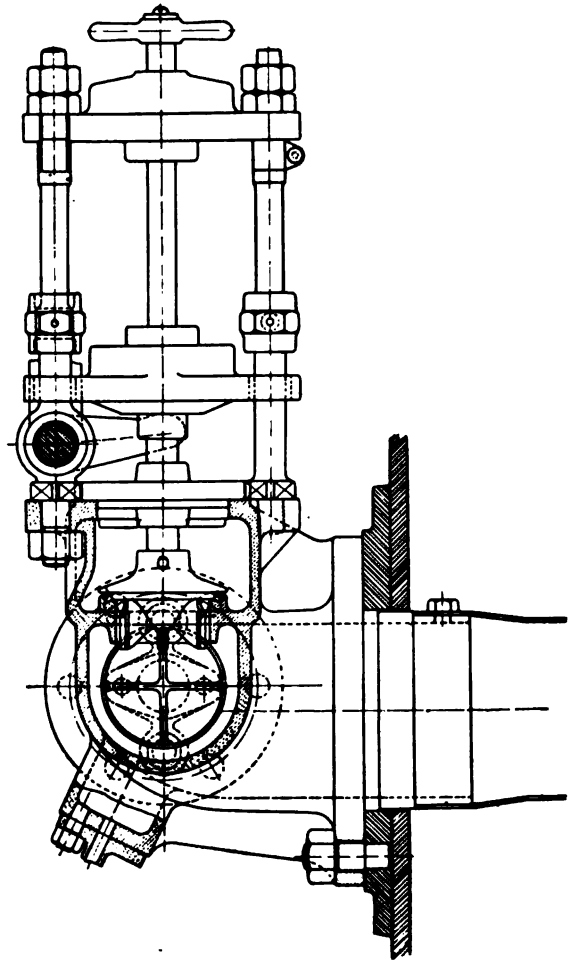


Fig. 495.

§ 325. **Steam Stop Valve.**—The internal diameter of the boiler stop valve, opening into the main steam pipe, is determined from the formula—

$$d = c\sqrt{\frac{H}{p}}$$

Where d = diameter of valve in inches.
H = heating surface of the boiler in square feet.
p = absolute boiler pressure in pounds per square inch.
c = 1·26 to 1·35 for boilers with natural draught.
c = 1·4 to 1·62 for boilers with forced draught.

The stop valves for the auxiliary steam pipe are of the same diameter as the pipes connected to them (see § 253). The valves should be as strong as possible, and the flanges joining them to the boiler should be made specially thick and stiffened with ribs.

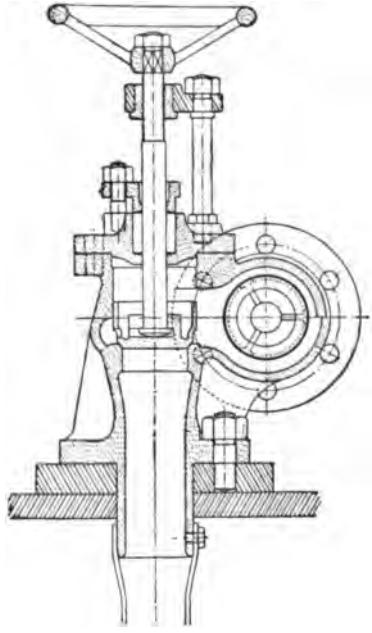


Fig. 496.

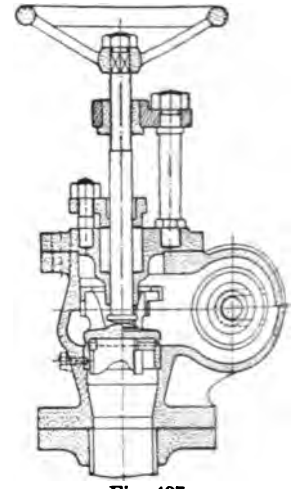


Fig. 497.

In other respects all that has been said concerning stop valves in § 241 holds good for these valves also.

If there is no steam dome to the boiler, it is advisable to have an *internal steam* collecting pipe at the top of the steam space, which connects with the main stop valve and the safety valve. The longitudinal or cross slits or holes on the upper side of the collecting pipe have a total sectional area about twice as large as that of the pipes connected to them. Copper or brass was the material formerly used for these pipes. In modern practice, copper, or alloys of copper, are no longer admissible for fittings inside the boiler, as they may set up corrosion, due to galvanic action. The internal steam pipes are therefore made of soft iron.

§ 326. **Feed Check Valves.**—In these the valve is not connected to the spindle, and the lift can be frequently adjusted by a screw spindle and hand wheel. A special stop valve is placed between the feed check valve and boiler, so that in case of need the latter may be entirely disconnected from the feed pipes. Both valves are made of bronze, and either fitted separately, or cast in one piece (see Figs. 496, 497). Where the valve is not connected to the spindle, separate seats should be fitted. Both the valve and spindle should be made as strong as possible, and the thread of the valve spindle should be outside the valve. Sometimes the main and auxiliary feed check valves are in one casting, which is fastened direct on to the boiler. In boilers working at moderate steam pressures (up to about 120 lb. per square inch) a three-way cock is often fitted, leading to both the feed check valves. To reduce the lift of the valve within the smallest limits, the diameter of the valve seat is made about ·16 to ·31 inch larger than that of the feed pipe connected to it (see § 256). The feed check valves must be as easy of access as possible. The pipe inside the boiler connected to the feed check valve is now generally made of iron, for the reasons given above, preferably of cast iron, because it is not so liable to rust, and is carried up as far as the lowest water level, where it is pierced with holes, to distribute the water as much as possible.

§ 327. **Water Gauges.**—These are generally tubular and made of glass, and have an external diameter of from $\frac{5}{8}$ to $\frac{3}{4}$ inch; but gauges with flat glasses (Klinger's patent) are also used. The top and bottom gauge cocks, or sometimes only the bottom one, are often fitted with self-closing valves (Leser, Dietze, Westphal, Burgemeister's patents, &c.), to prevent the firemen being scalded, should a glass break. The cocks are packed with asbestos. It is also necessary to protect the glass by some arrangement (such as wire netting, strong plate glass, &c.), so that, if the gauge glass breaks, the splinters may not injure any one. The length of the glass tubes between the nuts of the stuffing boxes is from about 10 to 14 inches.

In the Maas water gauge the glass is fixed into a separate holder or carrier, and the two can be taken out and fitted in together. The glass may thus be fixed to the holder at a distance from the boiler. The holder is secured to the heads of the gauge cocks by means of cones.

As a rule the gauges are not fitted direct to the boiler, but on to a *gauge column* of cast iron or brass, which is connected by copper tubes with the steam and water spaces respectively. The internal diameter of these pipes is from $\frac{3}{4}$ to 2 inches, according to the size of the boiler (the

Hamburg authorities require a diameter of at least 45 millimetres = say $1\frac{3}{4}$ inch), and special valves or cocks are often arranged so as to shut off the connection with the boiler. The height of the water gauge should be so arranged that the lowest working level of water is not more than 4 inches above the nut of the lower stuffing box, when screwed down. To mark it, an index should be fixed on the gauge column having a plate bearing the words—"Lowest water level." The gauge column is secured by wrought-iron brackets to the smoke box or boiler.

§ 328. **Pet-cocks or Valves.**—These may be fitted either direct on to the boiler, or if the conditions will not allow of this, they may be connected to the boiler by straight, thick, wrought-iron pipes. To admit of readily cleaning the valves or cocks and pipes from scale and sediment, they must have a straight way through them. The cocks are often asbestos packed to render them steam tight, and enable them to be easily worked. The cocks or valves may be screwed into the boiler, or preferably, attached to it by means of flanges and $\frac{5}{8}$ inch studs. If the pipes are of wrought iron, they are secured to the boiler with a fine thread and lock nut; a wrought-iron flange is generally fixed to the other end of the pipe, to which the pet-cock is fitted. Internal diameter of the pet-cock $\frac{1}{4}$ inch to $\frac{3}{8}$ inch. The lowest pet-cock is in line with the lowest water level; the upper cock is placed about 4 inches above it.

§ 329. **Density Cocks or Valves.**—A small cock or valve is fitted to the boiler, by means of which the attendant can draw off some of the water, and determine its density or percentage of salt. This valve or cock is so placed that it can be easily reached from the floor of the stokehold, and is generally made similar to the pet-cocks or valves on the gauge columns and pipes. Diameter from $\frac{1}{4}$ to $\frac{3}{8}$ inch.

§ 330. **Blow-off Cocks or Valves.**—In order to keep the pipe connections as short as possible, this valve or cock is placed close to the bottom of the boiler, and it and the spindle are usually made of gunmetal, the thread on the spindle being outside the valve. The valve must be so fitted that the pressure tends to lift the valve. The diameter of this valve, and of the pipe connected to it, is generally made rather less than that of the boiler feed pipe. The internal blow-off pipe is made of cast or wrought iron, copper, or brass, and terminates about $\frac{5}{8}$ to $\frac{3}{4}$ inch above the bottom of the boiler.

§ 331. **Scum Cock.**—This is also made of brass, and so fitted to the boiler that the pressure tends to lift the valve, the thread of the spindle being outside. Diameter of the valve is about 0·65 to 0·8 the diameter of the boiler feed pipe. The internal scum pipe or trough is carried along the lowest water level inside the boiler at about the middle of the surface of the water. The end of the pipe is funnelled or dish-shaped, to collect the scum.

§ 332. **Boiler-emptying Plug.**—A plug 1 to $1\frac{1}{2}$ inch diameter is sometimes fitted to the lowest part of the boiler. It consists of a gunmetal casting, bolted to a stiffening ring, and fitted with a screwed gunmetal plug.

§ 333. **Apparatus for Improving the Circulation of Water in the Boiler.**—In order to avoid unequal heating of the water and boiler in a cylindrical boiler when raising steam, and to prevent undue straining, which causes leakage, an apparatus for heating the water and producing circulation (known as a *hydrokineter*) is often used. It is made like an injector, and fitted in a suitable position inside the boiler. The steam to drive it is obtained from a donkey boiler. The apparatus sucks up the cold water from the bottom, heats it, and delivers it into the upper part of the boiler. It is sometimes fitted to the internal feed pipe of the boiler. The steam pipe to the hydrokineter is generally made from $\frac{3}{4}$ to 1 inch in diameter, and to prevent the water in the boiler from escaping, the pipe should have a non-return valve fitted to the boiler shell, in which the valve is not fixed to the spindle. The diameter of the steam nozzle is about $\frac{1}{8}$ to $\frac{1}{4}$ inch.

§ 334. **Summary of Remarks** in reference to marine boiler fittings. All fittings are usually secured with studs to wrought iron or cast steel stiffening rings riveted to the boiler shell; the studs must not penetrate the latter. The flanges to hold the different fittings should be made as strong as possible and strengthened with ribs, while the bolts securing them should, if possible, be $\frac{1}{8}$ inch thicker than the corresponding ordinary bolts. Bolts less than $\frac{5}{8}$ inch diameter should never be used. All fittings must have projections exactly fitting into, and passing through the boiler plate, to which the internal pipes can be secured.

§ 335. Regulations affecting Marine Boiler Fittings.

1. German Government Regulations.

Safety Valves.—All ship's boilers must have at least two safety valves. With the exception of ocean-going vessels, one of these valves must be

so placed that the prescribed load on it can be easily verified from the deck. The valves must be so arranged that they can be eased when necessary, and that they blow off the moment the blow-off pressure is reached in the boiler.

Feed Check Valves.—Every boiler must have a feed check valve, which is closed by the pressure of the water in the boiler when the feed is cut off.

Water Gauges.—The level of the top of the combustion chamber measured athwartships must be indicated on the casing or shell of every marine boiler in a clear and permanent manner; and two water gauges must be placed on this shell, in the same plane athwartships, symmetrical to the centre line of the boiler, and as far apart, right and left of it, as possible. There must also be a second arrangement for indicating the water level in the boiler. Each of these fittings must have a separate connection with the inside of the boiler, or if they have a common connection the gauge column must have an area of at least 60 square centimetres, say 9½ square inches. If pet-cocks are used, the lowest of them must be in the same plane as the lowest water level. All pet-cocks must be so arranged as to have a straight way through, for cleaning the pipe from scale and deposit.

The lowest permissible water level must be clearly indicated on the gauge column, and must be at least 4 inches above the top of the combustion chamber. This minimum distance must be maintained when the ship is inclined at an angle of 4° to the horizontal, in the boilers of river and lake steamers, and of 8° in the boilers of ocean-going vessels. This rule does not apply to water-tube boilers consisting of tubes over 4 inches diameter, nor to combustion chambers in which there is no danger of the plate in contact with the steam space becoming red hot. The danger may, as a rule, be considered as obviated if the wetted heating surface of the boiler is, with natural draught, at least twenty times as much, and with forced draught at least forty times as much, as the grate area.

Pressure Gauges.—Each boiler must have two pressure gauges, on which the highest permissible steam pressure must be marked. One of these must be easily visible to the firemen, and the other on deck at a place convenient for observation, except in ocean-going ships. If there are several boilers in the ship, connected up, it will be sufficient to have one pressure gauge on deck, in addition to the one fitted on each boiler.

Marking.—Each boiler must, in accordance with regulations, have the highest steam pressure, maker's name, factory number, year of com-

pletion, and the standard lowest water level marked on it in a clear and permanent manner. These particulars must be shown on a metal plate, fixed to the boiler with copper rivets in such a way that it is visible when the boiler is lagged.

Testing.—Every new boiler must, after it is put together, and before it is lagged, be tested hydraulically with all openings sealed. For boilers which are not intended for a pressure of more than 5 atmospheres, or 70 lb. per square inch above atmosphere, the test pressure to be double the working pressure; for all other boilers, the test pressure to be 5 atmospheres, or 70 lb. per square inch above the working pressure. One atmosphere is here equivalent to 1 kilogramme per square centimetre, or 14·22 lb. per square inch.

2. German Lloyd's Regulations.

Safety Valves.—Every boiler must have at least two safety valves, fitted directly to the boiler, with easing gear of sufficient lift. Suitable methods must be adopted, so that the valves cannot be loaded in excess of their specified pressure. The safety valve casings must be fitted with a drain pipe. Superheaters which can be disconnected from the main boilers must also be fitted with a safety valve.

Main Stop Valve.—Every boiler must be provided with a main stop valve, to shut off the main steam piping. If there are several boilers, and only one superheater, the latter must be so arranged that it can be shut off from the main steam piping.

Feed Check Valves.—Every boiler must have two feed check valves, which can be regulated by means of an outside screw; they must be fitted direct to the boiler, and arranged to serve the two feed pipes. The pressure in the boiler should tend to keep the valves shut, and they must be arranged so that they can be examined without having to empty the boiler.

Blow-off Valves or cocks must be fitted direct to the boiler.

Water Gauges.—Two of these, with an index showing the lowest water level, must be provided for each boiler. They must be in the same plane athwartships, at the same height, and symmetrical to the centre line of the boiler, and be clearly visible. The internal diameter of the pipes connecting the steam and water spaces must not be less than $1\frac{3}{8}$ inch in boilers having 320 square feet of heating surface, or under, and in larger boilers not less than $1\frac{3}{4}$ inch. Below the mark indicating lowest water level there should be at least 4 inches of gauge glass. With double-ended boilers there should be two gauge glasses

at one end, as already mentioned, and one in the middle at the other end. It must be possible to shut off the gauges easily from the boiler. Besides these water gauges, there should also be test-cocks or some other suitable arrangement for ascertaining the water level.

Pressure Gauges.—Every boiler must be provided with two pressure gauges, upon which the working pressure is marked in red; one should be in view of the firemen, the other in a prominent and easily accessible position in the engine-room. Suitable connections must be provided, to which the test gauges can be fitted.

Marking the Boiler.—Every boiler must have the highest allowable steam pressure, maker's name, factory number, year of completion, and standard lowest water level marked on it in a clear and permanent manner. These particulars must be shown on a metal plate, fixed to the boiler with copper rivets in such a way that it is visible after the boiler has been lagged.

Test Pressure.—Boilers which are intended for a working pressure of not more than 70 lb. per square inch are tested for double the maximum working pressure (above atmosphere); all other boilers are subjected to a test exceeding the working pressure by 70 lb. per square inch.

3. British Lloyd's.

Safety Valves.—Every boiler is to be provided with two *safety valves*, loaded to the working pressure in the presence of the surveyor. If the working pressure is more than 60 lb. per square inch, the valves must be loaded to 5 lb. above the normal working pressure. If ordinary valves are used, their total area is to be at least $\frac{1}{2}$ square inch to each square foot of grate surface. If any special forms of valve are used, they must be tested under steam in the presence of the surveyor; in no case must the steam pressure rise more than 10 $^{0}/_{0}$ above the working pressure. An approved safety valve is also to be fitted to the superheater. Donkey boilers may be fitted with one safety valve only, if the area of the latter is at least $\frac{1}{2}$ square inch for every square foot of grate area. Every safety valve must be so arranged that it cannot be overloaded when steam is up, and it must be provided with suitable easing gear which must lift the valve itself. All valve spindles must extend through the cover of the valve casing, and be fitted with a socket and cross handles, allowing the valves to be raised, turned round on their seats, and tested to see that they are in proper working order, at any time.

Stop Valve.—Each boiler must be fitted with a stop valve, so that it can be worked independently.

Blow-off Cocks.—Every boiler must be fitted with a blow-off cock, which is independent of the blow-off cock fitted to the ship's side. Every boiler must be provided with an accurate *pressure gauge.*

Test Pressure.—The boilers must be tested by hydraulic pressure up to double the working pressure in the presence of Lloyd's surveyors, and carefully gauged while under test.

4. Bureau Veritas.

Safety Valves.—Each main boiler must have at least two safety valves of approved design. Their total area must be such that with not less than twenty minutes' hard firing the pressure on the safety valve does not exceed the working pressure by more than 10 °/₀. If the boiler is worked with forced draught, the sectional area of the safety valves must be increased to correspond with the higher evaporative power of the boiler. Arrangements must be made and gear provided to ease the safety valves from the deck, or from the floor of the stokehole.

Main Stop Valve.—If there are several boilers in the ship, they must be capable of being worked together or independently; for this purpose stop valves must be provided between the boilers and the common superheater, or between the different superheaters and the main steam pipe. The steam piping to the auxiliary engines must be independent of the main steam piping, so that the main engine may not be affected when the winches and pumps are worked. There should be an arrangement for shutting off the boiler stop valve, either from the floor of the stokehold or from the deck.

Blow-off Cocks.—The blow-off pipe must be so arranged that one cock is fitted direct to the outside of the boiler shell, and one to the ship's plating, and the scum cock must be fitted in a similar manner.

Water Gauges.—Every boiler must be fitted with at least two water gauges, either with one gauge glass and one set of test-cocks, or with two gauge glasses. Boilers fired at both ends must have a similar arrangement at each end.

Every boiler must be fitted with at least one *pressure gauge.*

PART VI.

MEASURING INSTRUMENTS.

§ 336. **Pressure Gauges.**—There are two principal types of gauges used for marine purposes, viz., *flat-spring gauges* (Fig. 498) and *round-spring gauges* (Fig. 500) (Bourdon gauges). The principle underlying both types is based upon the phenomenon that the deflection of a spring subjected to a moderate strain is directly proportional to the force acting on it. The deflection of these springs is indicated on a suitable dial, the scale of which is determined experimentally by means of a mercury column open to the air. As the elasticity of the springs diminishes with increase of temperature, the pressure gauges must not be brought in contact with hot walls or hot steam. Therefore the makers stipulate that the instruments must only be exposed to moderate heat, and protected from the hot steam by a small *column of water* (Figs. 499, 501).

Every pressure gauge must be fitted with a stop-cock (usually a three-way cock), to shut off the steam pressure or to throttle it if necessary, should the spring be exposed to violent fluctuations of pressure in the steam space. For boiler pressure gauges, a flange (Fig. 502) must be provided, to which the test gauge can be attached. These gauges are generally made with double pointers and springs, on the Bourdon principle; they are extremely sensitive and very accurately adjusted. Wherever an engine plant of any size is at work, these gauges should always be to hand, for continually checking the working gauges.

§ 337. **Thermometers.**—For low temperatures, up to about 400° Fahr., *ordinary* mercury thermometers are used, and should be surrounded with a casing, to protect them from injury. When it is desired to take the temperature of steam in piping, &c., under pressure, stuffing boxes and other protections should be provided, to prevent the thermometers from being blown out by the pressure. As even with the most carefully constructed thermometers there is always a risk of their breaking, it is advisable, in order to obviate this danger, to protect them by a solid casing or cup screwed into the steam pipe, &c. (see Fig. 503). The heat is more effectually transferred if this cup is filled with mercury or oil. It may also be filled with fine iron filings, which can be extracted from the cup by means of a magnet. It should be noted that quick-

silver is unsuitable for use with brass cups, on account of its action on the brass. It is often contended that enclosed thermometers are inaccurate, but this objection does not apply in practice, as for the degree of accuracy required in these measurements, any slight injurious effect of the casing may be neglected.

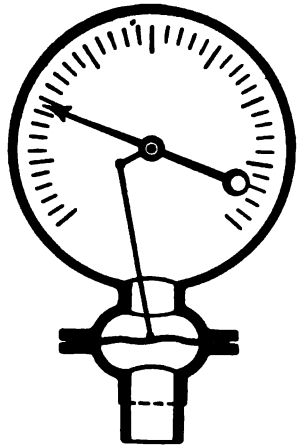


Fig. 498.

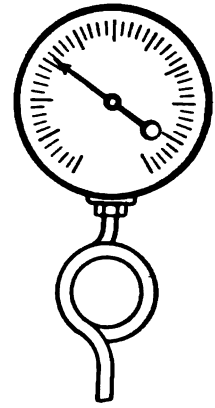


Fig. 499.

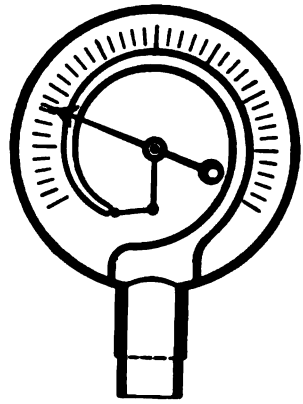


Fig. 500.

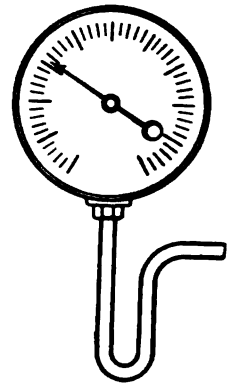


Fig. 501.

If the temperatures to be measured approach the boiling point of mercury (648° Fahr.), the ordinary mercury thermometers cannot be employed. In this case, mercury thermometers partly filled with nitrogen are used; *i.e.*, thermometers in which the top of the glass tube is widened, and filled with compressed nitrogen gas, to prevent the

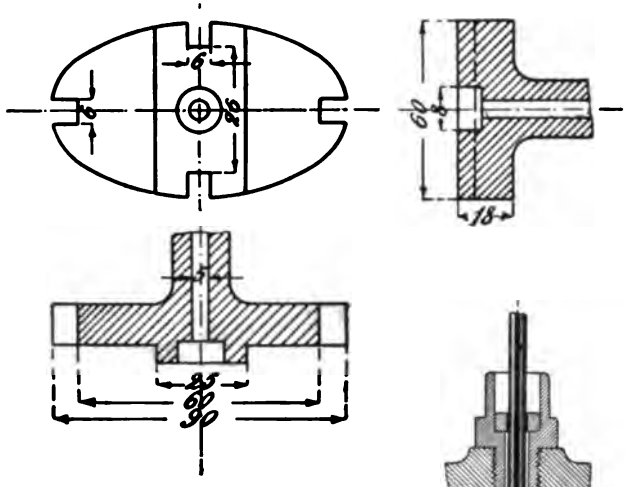


Fig. 502.

Fig. 503.

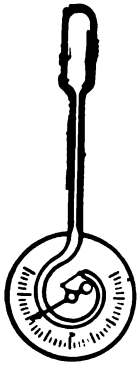


Fig. 504.

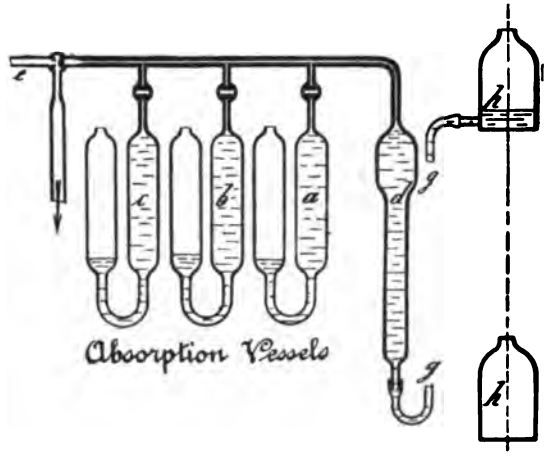


Fig. 505.

mercury boiling. *Pressure thermometers*, based on the principle that at certain pressures saturated vapours acquire well-defined temperatures, may be usefully employed. These thermometers, called "*mercurial pyrometers*," are made in the following way:—At the bottom is a flat wide vessel (the evaporator), and to this a glass tube, as long as may be required, is attached, carrying at one end the pressure gauge, with lever and pointer (Fig. 504). The tube is filled with ether, water, or quicksilver, according to the height of the temperature to be measured. As long as these instruments are used to indicate temperatures below red heat, *i.e.*, up to about 1,300° Fahr., they are perfectly reliable. Above that, they must be often recalibrated, by means of standard air thermometers. *Graphite pyrometers* are much used for commercial purposes, but are not suitable for marine work, as they are too inaccurate, if constantly employed.

§ 338. **Analysis of the Flue Gases.**—The determination of the excess of air is a valuable aid in judging of the efficiency of combustion, and this may be best effected on board ship by the use of the *Orsat apparatus* (Fig. 505). It is generally arranged to test the gases for their percentage of carbon dioxide, carbon monoxide, and oxygen, and indirectly of nitrogen. Three absorption bottles, *a*, *b*, and *c*, are each connected by a cock to a glass suction or capillary tube, bottle *a* being filled with solution of caustic potash, *b* with pyrogallic acid, and *c* with ammoniated solution of cuprous chloride. At one end of the capillary tube is a three-way cock, which has a connection on one side to the smoke box, on the other to a water jet pump. To the other end the measuring burette *d* is attached, which has a capacity of 100 cubic centimetres, and is carefully marked to scale in centimetres. At the lower end of the burette is a rubber tube *g*, which forms the connection with a levelling bottle *h* filled with water.

The gases are analysed as follows:—The absorption fluids are first introduced into that half of one of the double bottles which is nearest the cock, and the cocks closed. The levelling bottle is raised till the measuring burette is filled with water up to the top, and the three-way cock is turned to open communication between the smoke box and the jet pump. By means of the latter, all the air is sucked out of the connecting pipe, and it is filled with flue gases only. The three-way cock then connects the smoke box with the measuring burette, and the levelling bottle is lowered, till there are exactly 100 cubic centimetres of gases in the burette, and the cock shut off. The cock to *a* is then opened, and by raising the levelling bottle the gases are passed over into the bottle containing caustic potash, for absorbing the CO_2. The surface of the reagent is increased by putting little tubes of glass into

the bottle. Absorption being complete, the levelling bottle is lowered, the rest of the gases passed back into the burette, the cock at *a* closed, and the volume of the remaining gases read off. The difference in the readings gives the percentage by volume of carbonic dioxide in the flue gases. By opening the other cocks at *b* and *c*, one after the other, the gases are passed successively into the other bottles, the oxygen and carbonic monoxide are absorbed, and the final residuum is nitrogen. Great accuracy is required in making these analyses, and special care must be taken that all the cocks and connections are perfectly tight. The order given above in which the gases are absorbed by the different reagents must be strictly adhered to.

§ 339. **Draught Gauge.**—The draught is measured in inches of water. For this purpose the best and simplest apparatus is a glass tube bent in the shape of a U, Fig. 506 (U gauge). In practice the mistake is often made of not stating where the draught was measured. To make the measurements of greater value, it should be mentioned, not where the vacuum was measured, but between which parts of the boiler, for instance between the ash-pit and combustion chamber, or between the smoke box and the outer air.

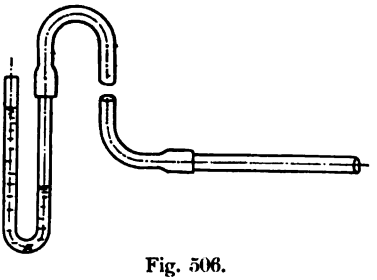


Fig. 506.

§ 340. **Determination of the Heating Value of Coal.**—The better kinds of coal develop, more or less, the same quantity of heat during combustion (have the same calorific value), nevertheless it is sometimes useful to determine experimentally the heat efficiency of a given fuel. It is not advisable that the engineer should himself undertake these experiments (by burning the coal with oxygen in a Mahler bomb), because they require much practice and occupy considerable time; it is better to have the coal tested in a chemical laboratory, but great care must be exercised in selecting the samples. To get a representative sample of the coal, a shovelful is taken from different parts of the coal of which it is desired to ascertain the calorific value, carefully crushed and then well mixed. From different parts of this heap small samples are again taken, and further mixed, and the process is continued until a small representative sample is obtained, which is hermetically sealed, and forwarded for testing. As the percentage of water contained in the coal materially affects the results, it should be stated whether the samples are taken from wet or dry coal.

§ 341. **Determination of the Amount of Moisture in the Steam: Dryness Fraction.**—The best apparatus yet introduced for this purpose is Peabody's throttling calorimeter (see Fig. 507). This instrument does not require complicated arrangements, or occupy much time in manipulation. The apparatus consists of a vessel of about 120 cubic inches capacity, which is connected through a valve with the steam pipe and the condenser respectively, and is carefully protected from loss of heat by radiation. A pressure gauge is fitted in the steam pipe, and a

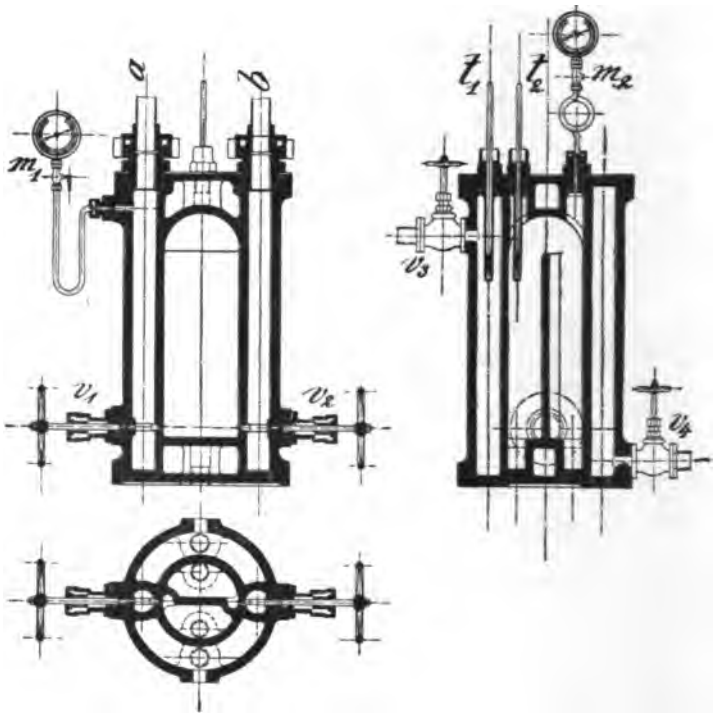


Fig. 507.

thermometer and pressure gauge are fixed in the calorimeter. The valve v_1 is so adjusted that the steam in the calorimeter is much throttled, and is thus at a considerably lower pressure than in the steam pipe. If the absolute pressure in the steam pipe is p_1, then the heat of vaporisation q_1 and the latent heat r_1 corresponding to this pressure, can be ascertained from the Steam Tables, and the values q_2 and r_2 corresponding to the pressure p_2 of the throttled steam can be ascertained in the same way. As, for every pound of mixture in the steam pipe, there will be x_1 lb. of

steam, and $(1 - x_1)$ lb. water, the heat contained in 1 lb. of steam will be $Q_1 = q_1 + x_1 r_1$. As soon as the steam in the calorimeter comes to rest, it will contain, except for any minute losses from radiation, the same amount of heat as when in the steam pipe. This will consist of $Q_2 = q_2 + r_2 + c\,(t'_2 - t_2)$, c being the specific heat of dry saturated steam expanding at constant pressure (about 0·48), t'_2 the temperature shown in the calorimeter, and t_2 the temperature of saturation corresponding

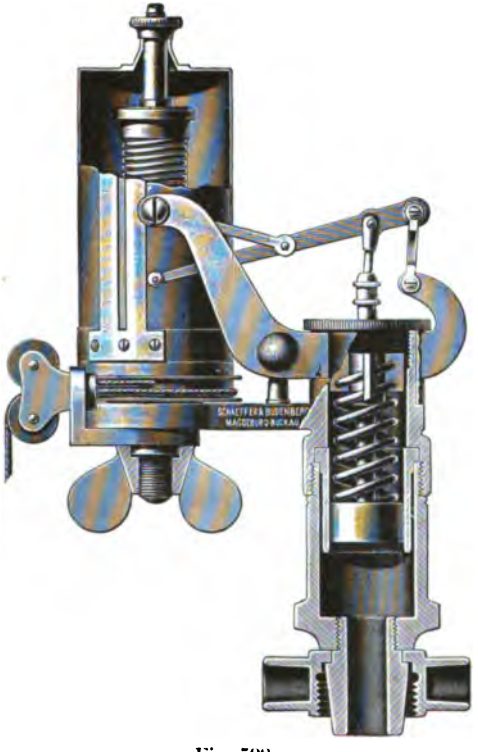


Fig. 508.

to the pressure p_2. The loss from radiation being negligible, $Q_1 = Q_2$, that is, $q_1 + x_1 r_1 = q_2 + r_2 + c(t'_2 - t_2)$, from which x_1 can be calculated.

These determinations will only yield reliable results if the apparatus used is very accurate, and the percentage of water in the steam does not exceed 5 °/₀. If the steam is supposed to contain more than 5 °/₀ of water, a separator must be fixed just in front of the calorimeter, and the results obtained from it added to those of the calorimeter. To avoid any radiation of heat into the surrounding air, the jacket round the

separator is so heated that the thermometer in it registers exactly the same temperature as the one in the calorimeter.

§ 342. **Indicators and their use.**—Those most generally employed in Germany are—

1. *Schäffer & Budenberg.*
2. *Dreyer, Rosenkranz, & Droop.*
3. *Crosby.*

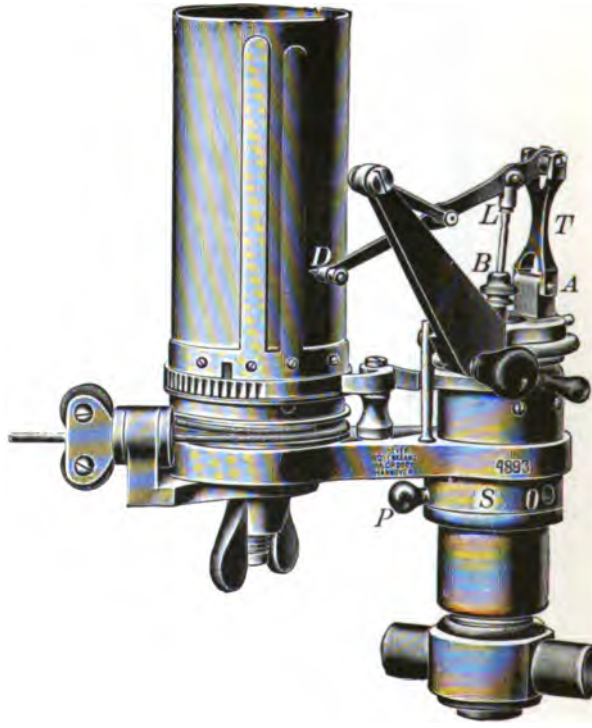


Fig. 509.

Figs. 508 to 510 show the latest types made by these firms. In all the instruments the compression of a spring adjusted to scale is utilised to represent the varying pressure of the steam during one stroke of the engine. The deflections of the spring, proportionally enlarged by the parallel motion of the pencil, are transferred to the paper, and the manufacture of the instrument has now been brought to such perfection,

that the scale of the spring is a constant throughout the height of the diagram.

§ 343. **Study of the Indicator and its Accessories: Preparations for indicating.**—1. Remove the paper cylinder and lubricate the axis or pivot of the drum. The tension of the drum spring must be such, that it is sure to secure the return motion of the

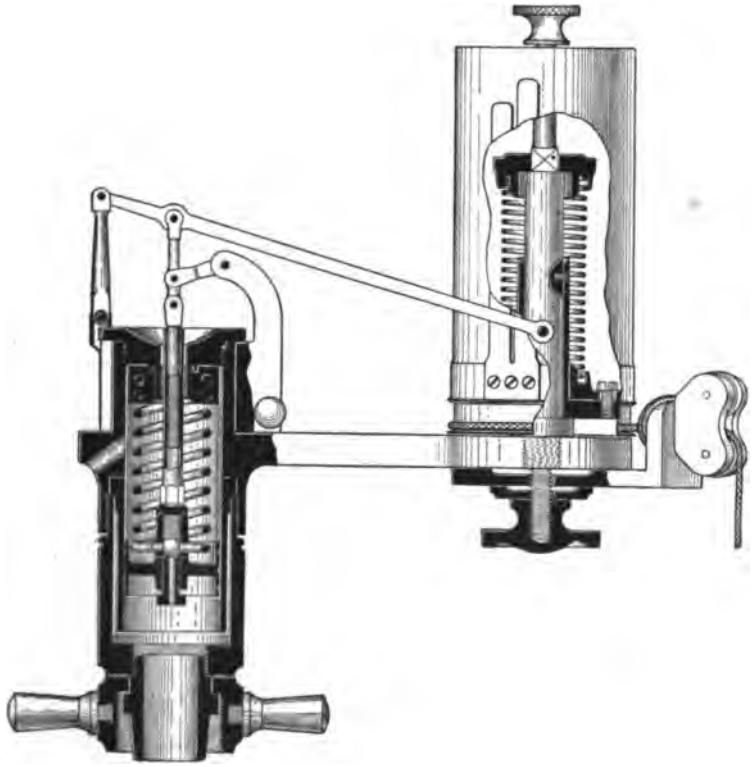


Fig. 510.

drum. The tension must therefore be greater if the number of revolutions is high. Too great a tension is, however, to be avoided.

2. Remove the pencil and piston attached (according to the instructions given with each instrument). The inside of the indicator cylinder must be perfectly free from any flaws, irregularities, or dirt, and must be cleaned if necessary with soft wadding and oil; any hard or carbonised

oil must be removed with petroleum. The use of sharp or pointed tools must be carefully avoided, otherwise the bore of the cylinder may be affected, and the piston no longer fit accurately; if repairs are required they should only be done by skilled hands.

3. *To disconnect the Piston, Piston-rod and Springs.*—The screws after they have once been loosened with a key must work easily, and be cleaned if necessary with petroleum, as oil deposit, especially when indicating high pressures, makes the instrument work stiffly. The pencil and paper should be tested while the engine is running light (this precaution is often neglected). Little can be altered at this stage, but the result of this test shows whether the instrument can be used or not for a given experiment.

4. *To adjust the Spring.*—Before this is done, it should be known between what limits of pressure the instrument is required to indicate. If the spring is too weak, the indicator piston will hit the cylinder top, and an imperfect diagram will be drawn; if it is too strong, the top of the diagram will not be fully developed, and the area will be too small. The springs are calibrated and marked with a number or fraction which represents the distance they compress in inches, per pound per square inch of steam pressure.

5. *Pencils.*—Metal pencils are generally supplied with the instruments, with prepared paper, on which the diagrams are drawn. It is often better, however, to use ordinary smooth writing paper and a common lead pencil, as the diagrams thus drawn are clearer. This is especially the case when several diagrams have to be taken one over the other, the various lines being more sharply defined.

6. *To fit the Indicator on to the Engine.*—This is done without putting in the indicator piston and pencil. During the interval before indicating the engine, the indicator cocks must be put in and tested, to see if the threads of the cock and indicator fit each other, if the plugs of the cocks work easily, and if they are properly adjusted.*

7. An indicator table should be drawn up, showing at a glance what indicator springs are to be used for the different cylinders, what make of indicator is used for the top and bottom of each cylinder, and any errors to be allowed for, as for example :—

* It often happens that the plug is so fitted in that instead of being able to draw the atmospheric line, steam blows through. If the washer cannot be turned through 90°, the plug itself must then be turned through 180°, and this can only be done when steam is shut off.

	H.P. Cylinder.		No. 1. M.P. Cylinder.		No. 2. M.P. Cylinder.		L.P. Cylinder.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Make and Number of Indicator	Schäffer & Budenberg No. 648	Schäffer & Budenberg No. 649	Crosby No. 483	Crosby No. 487	Crosby No. 486	Crosby No. 485	Dreyer & Rosenkranz No. 590	Dreyer & Rosenkranz No. 591
Scale of Spring.	$\frac{1}{100}$ inch per lb.	$\frac{1}{100}$ inch per lb.	$\frac{1}{50}$ inch per lb.	$\frac{1}{50}$ inch per lb.	$\frac{1}{50}$ inch per lb.	$\frac{1}{50}$ inch per lb.	$\frac{1}{8}$ inch per lb.	$\frac{1}{8}$ inch per lb.
Remarks	Steam jacketed	Steam jacketed	Pencil not working	Spring newly tested	—	—	Aluminium drum	Aluminium drum

8. Before putting in the pencil and paper, steam should be well blown through the indicators, to clean and heat the passages. On the low-pressure cylinder it is advisable to blow through only during the period of admission, to prevent cold air being drawn in. Immediately before setting the indicator, the paper and pencil should be tested, and the piston and spring examined to see if they are properly and firmly screwed up, and whether the spring corresponds with the scale given in the table. The piston is then dipped in thick cylinder oil, well smeared, and fitted in. The precaution of keeping the pencil and its gear free to move on the indicator cylinder is often overlooked, but this is important, to avoid any hitch while taking a diagram.

§ 344. **The Driving Gear.**—The principal point to be kept in view is that the paper and drum must move in proportion to the stroke of the piston. To effect this a lever (Fig. 511), pivoted at a short distance from one end, is almost always used. For other methods of driving the indicator see Figs. 514, 515. It may be worked in one of two ways, either from the crosshead, or by an eccentric on the shaft. In the former case the indicator cord must be at right angles to the central position of the driving lever (see Figs. 512 and 513); in the latter it must be so arranged, that the eccentric and the engine piston are at the top of their respective strokes at the same moment.

Fig. 515 shows a method of driving the indicator in high-speed engines. The link motion must never have any play or backlash, and the cord must not be too heavy or the bends too sharp, otherwise the diagram will be distorted. The end of the driving lever, on to which the cord is hooked, must be free from sharp edges and corners, to avoid any risk of injury to the operator. After putting on the cord the

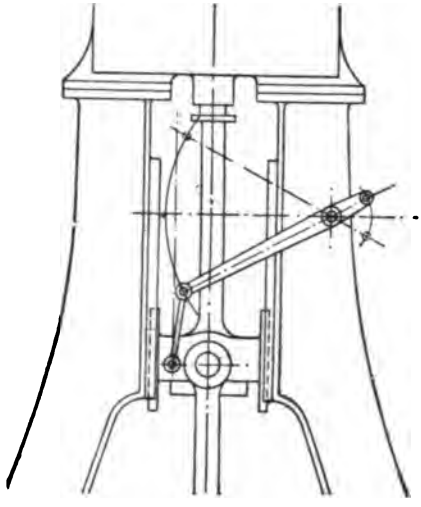


Fig. 511.

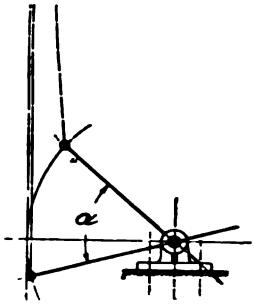


Fig. 512.

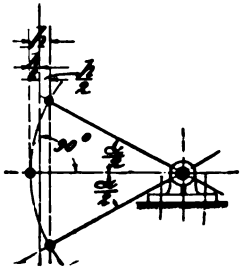


Fig. 513.

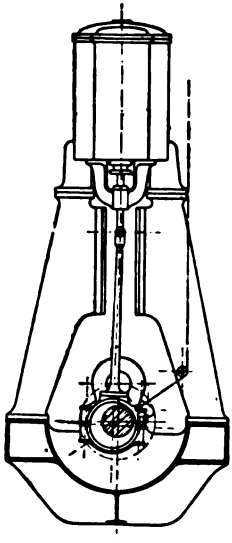


Fig. 514.

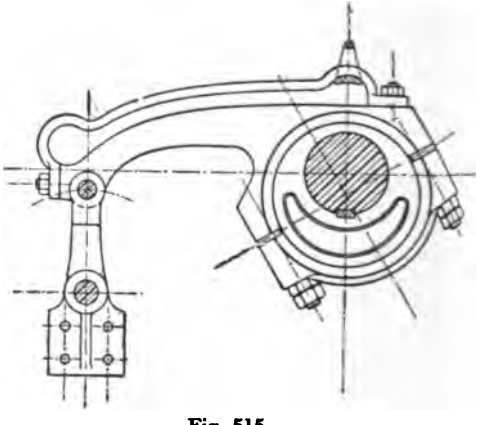


Fig. 515.

drum must be examined to see if the diagram can be properly drawn in its highest and lowest positions of the crank, or whether there is any defect in guiding the cord.

§ 345. **Putting on the Paper.**—The paper must lie smoothly all over the surface of the drum, otherwise errors will occur in drawing the diagram, and the paper may get torn. If necessary the clips must be removed and readjusted.

§ 346. **Planimeter.**—To work out the diagrams the *Amsler polar planimeter* is generally used. It consists of a movable arm *cd*, and a fixed arm *ab*, Fig. 516. If the instrument is in proper adjustment, and the pencil *d* is made to trace exactly the outline of the diagram, starting from any given point, the reading on the wheel, multiplied by the constant for the apparatus, gives the area of the diagram in units corresponding to the constant. There is usually a sleeve by which the fixed arm can be shifted along the movable arm, and the planimeter adjusted to the scale of the diagram. In this case the reading gives the mean height of the diagram in a definite unit of measurement. In using the planimeter it is always advisable first to draw the vertical tangents to the diagram (at right angles to the atmospheric line), in order to mark off exactly the length of the diagram upon the atmospheric line; the process of placing the diagram in position is then quite easy. It should be noted, however, that there are always slight variations in the lengths of diagrams, and the distance between the extreme points should be checked for each diagram. It is also necessary to check the constants of the planimeter, and if they are not known they must be determined, by accurately drawing a number of rectangles of known size, and then measuring them by the planimeter. The calculated area of each rectangle divided by the planimeter reading gives the constant.

To find the constants forming the basis of a diagram already plotted, all the rectangles are drawn with the same base line, and the planimeter traced over them; in this case the height measured, divided by the reading, gives the constant required. Care must be taken when using the planimeter that the wheel runs easily over smooth and uncreased paper. To begin measuring the diagram, the movable arm is placed at a tangent to the diagram, and at the same time at right angles to the fixed arm. In this position, errors in plotting at the beginning and end of the measurement have least effect on the result. The edge of the wheel must be carefully protected from injury.

§ 347. **Schlick's Pallograph.**—To determine the vibrations in the hull of a ship, the Schlick pallograph has been much used of late.

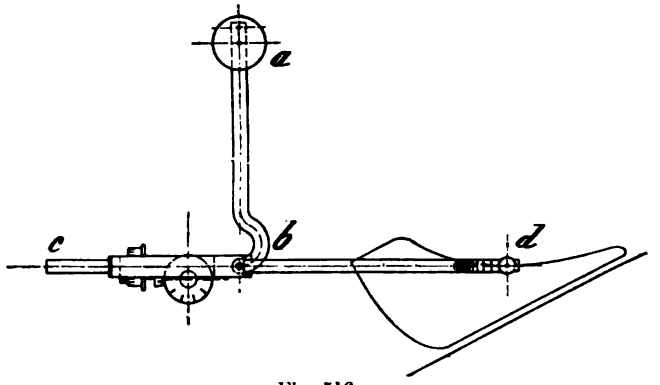


Fig. 516.

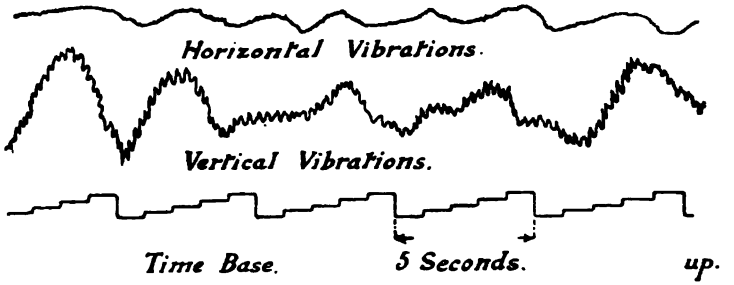


Fig. 517.

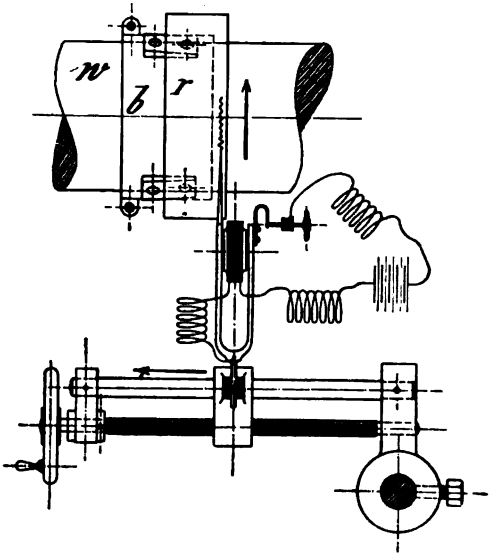


Fig. 518.

This instrument mainly consists of four parts, a strip of paper driven by clockwork, and three pencils, one of which records the vertical vibration, the second the horizontal, and the third marks the time base. The vibrations are rendered visible by heavy weights suspended from springs, and connected to the two first-named pencils. The vertical vibrations are recorded by a weight which can only swing vertically relative to the point of suspension, and the horizontal by a second weight which can only swing horizontally. The principle of the apparatus is based on the phenomenon that heavy masses are not susceptible to sudden changes of speed. Therefore the weights suspended from springs, and the pencils, do not take part in the comparatively rapid successive vibrations of the hull of the ship, but they are followed by the frame of the apparatus and the strip of paper, because these are fixed to the ship. The pencil for the time diagram is moved regularly to and fro by clockwork, and draws a stepped line upon the paper. Fig. 517 shows a diagram from this apparatus, from which the character of the vibrations can be clearly followed.

§ 348. **Instrument for Measuring the Uniformity in the Turning Moment of an Engine.**—A useful instrument for testing whether the turning moment of a marine engine is uniform is shown in Fig. 518. A collar *b* with four cotters is placed on the shaft *w*. Upon it a metal hoop *r*, carrying a strip of paper round it, can be fixed by sliding it along parallel to the axis of the shaft. A tuning fork controlled by an electro-magnet is made to vibrate strongly and uniformly by means of an interrupter, and the vibrations are transferred by a pencil to the blackened paper strip. While the engine is running, the tuning fork is carried along the shaft by means of a screwed spindle, and thus draws a continuous waved line on the paper. After the experiment the strip of paper is taken off, and from the variations in the lengths of the undulations the uniformity of the turning moment can be determined.*

§ 349. **Fottinger's Torsion Indicator** is used to register automatically by means of a curve the amount of the actual turning moment, *i.e.*, the actual tangential pressures during one revolution. The diagram of the torsional angles and curves gives respectively, on a different scale, the actual turning moment, as the twist of the shaft is exactly proportional to the effective turning moment.

The apparatus, Fig. 519, consists of a tube in two parts which is passed over the shaft, the forward end of the tube being firmly secured to it. The after free end carries a disc marked 1, and opposite to

* Compare *Journal of the Schiffbautechnischen Gesellschaft*, vol. i.

the latter a similar disc marked II is fixed on the shaft. Two points on discs I and II which, if the shaft is not in tension, are opposite each other, will, on account of the twisting of the shaft when running, be shifted along a short distance *s*, which can easily be calculated from the turning moment at any given instant, &c., from § 3. This slight movement (about ·06 to ·08 in.) of disc I in reference to II, is enlarged from 18 to 28 times, by means of the arrangement of levers (shown at Fig. 519). The tie rod *a b* is connected with disc I by the screw *k*, and moves the shorter arm of the bellcrank lever *b c d*, the axis of which *c* is fixed to disc II. To the longer arm *c d*, the rod *d e* is attached at *d*, and works the indicator lever *f e g*. This is capable of angular movement round a point *f*, which is fixed to disc II. The other end *g* of this lever carries the pencil for drawing the diagrams, the point of which faces towards

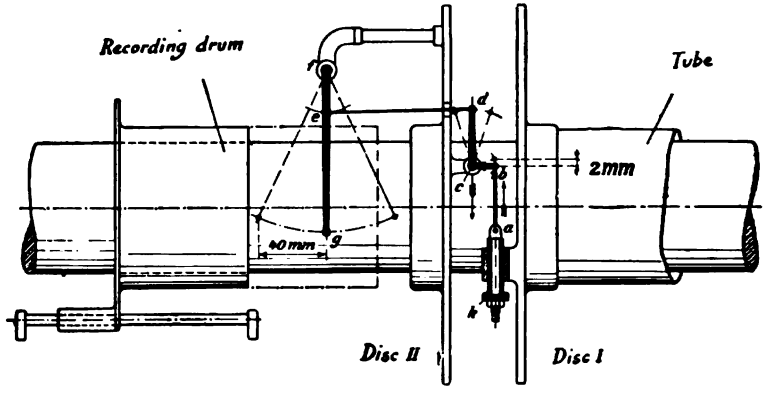


Fig. 519.

the centre of the shaft. When working, *g* swings from its point of rest in the direction of the axis of the shaft, in proportion to the turning moment at that instant, and all the levers, discs, tube, &c., rotate with the shaft.

Between the pencil and the shaft, and concentric with the latter, is a drum which is pushed axially along a slide between them, and upon this as many torsional diagrams as may be desired can be drawn. Paper is stretched round the drum, and upon it the pencil draws the curve proportional to the torsion, *i.e.*, the actual turning moment. Another pencil, not shown in Fig. 519, draws the datum line. To take off the diagram, the drum is pushed to the left, and a fresh paper can then be fitted to it.

Fig. 520 shows the construction of the first type of this instrument. The open end of the measuring tube is centred to the shaft by four

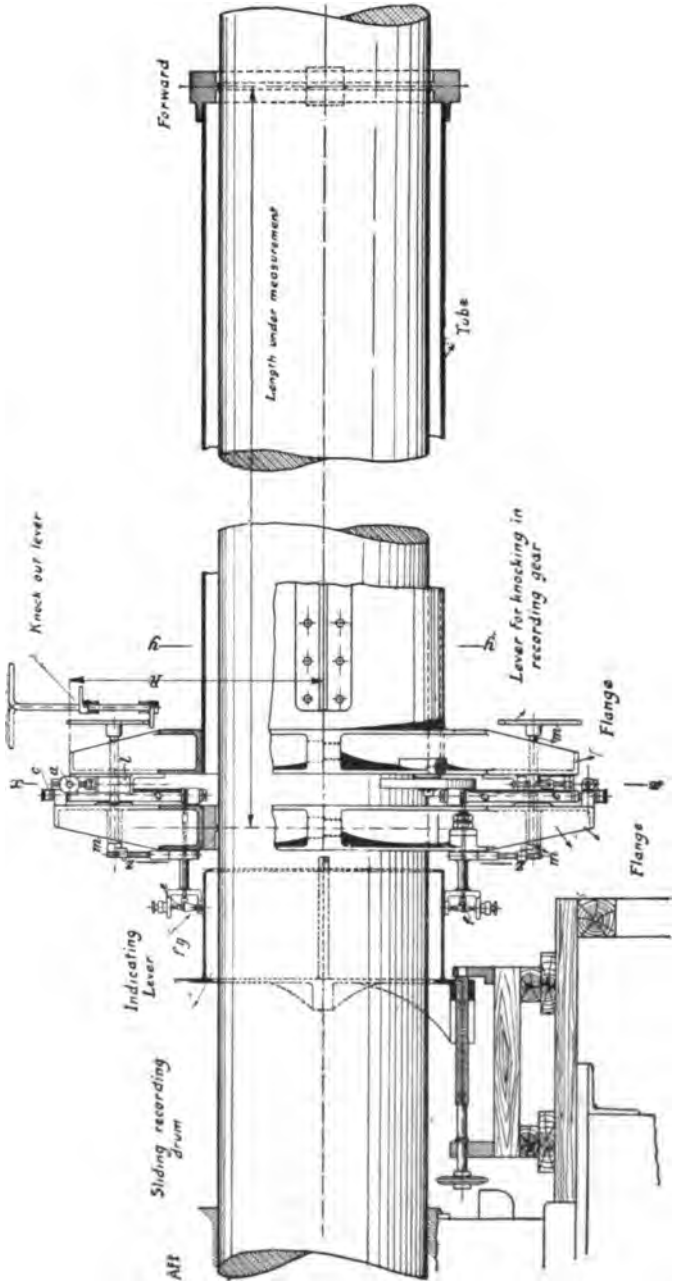

Forward
Length under measurement
Tube
Knock out lever
Lever for knocking in recording gear
Flange
Flange
Indicating Lever
Sliding recording drum
Aft

Fig. 520.

large adjustable rollers. The levers and their machinery are made in duplicate (and fixed to two opposite points) in case one of the pencils should fail to act. By means of the bellcrank lever *m m*, and a stop fixed to the body of the ship, the pencils can be raised from the paper, and the levers thrown out of action at the same time. Before starting the instrument, the zero point of the movable pencil is determined by turning the shaft backwards and forwards, and the stationary pencil is fixed in the position thus given. The actual work of the engine is found by measuring the diagrams with a planimeter. The apparatus registers automatically the variations in turning moment produced in a shaft of a given length. With a somewhat modified torsional indicator of this type, the B.H.P. of one of the 20,000 H.P. engines of the fast steamer "Kaiser Wilhelm II." was determined.

With another arrangement of the apparatus, suited to steam turbines and electric motors, the mean turning moment can be read off to scale on a stationary indicator.*

* See page 441, *Jahrbuch der Schiffbautechnischen Gesellschaft.*

PART VII.

VARIOUS DETAILS.

VARIOUS DETAILS.

§ 350. Bolts, Nuts, and Screw Threads, &c.

The allowable stress upon small screws is always taken as much less in proportion to that upon large screws, as the former are apt to be easily broken when being screwed in. This must be specially taken into account with bronze screws, which offer a smaller resistance to twisting. For dimensions of screw bolts with Whitworth thread, see Table No. 70.

In all screws over $\frac{5}{8}$ inch diameter, the tensile strength of which has to be as large as possible, such as bolts for connecting rods, crossheads, valve rods, &c., a fine thread is used (see Table No. 71). The heads of bolts that must be kept from turning, or which cannot be held fast while the nuts are being screwed on, must be fitted with a projection or feather (see Figs. 521, 522). A good sized fillet must always unite the head to the turned part of the bolt. When square heads are used, each side of the square must be made the same width as the corresponding hexagonal nut across the flats. Tap bolts or studs should be screwed into cast iron to a depth of one and a half times the diameter of the bolt; but if necessary the depth may be decreased to one and a quarter times the diameter. If they are to be screwed into tight-fitting holes, they must have a groove running lengthwise across the thread, to allow the air to escape while the bolt is being screwed home. Bolts and pins, the nuts of which have constantly to be taken off, should have the top rounded off, and their ends turned down for a short distance to the bottom of the thread to allow of the nuts being easily screwed on and off. This is especially necessary in the case of large horizontal screws, such as the screws for propeller nuts.

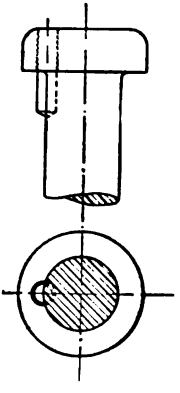


Fig. 521.

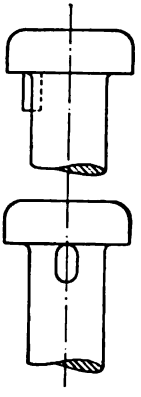


Fig. 522.

Iron or steel studs must not be screwed into bronze, as the iron rusts rapidly, especially if it is exposed to the action of sea water. The

heads of countersunk screws should not have a slit (Fig. 523), as turning the screws burrs the slits, and injures the countersinking. It is better to use a small square-headed screw (Fig. 524). In screws which have to be constantly taken out, it is advisable to case-harden the heads. Steel bolts or pins should have wrought-iron nuts, which do not wear the thread of the bolt so quickly as steel nuts. Nuts which have to be constantly taken off must be case-hardened if they are to retain their hexagonal shape.

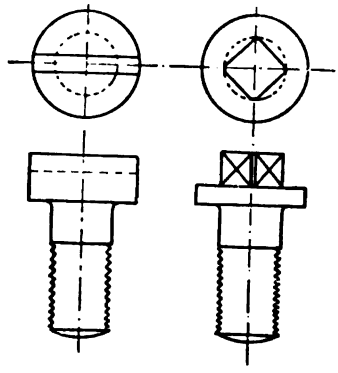


Fig. 523. Fig. 524.

Lock Nuts.—For small bolts separate lock nuts are generally used, for the depth of which see Table No. 70, or for ring or "Penn" nuts see Table No. 72. Split pins must not be considered sufficient to prevent the nut slacking back, but only to keep it from falling right off. For particulars and the methods of securing the nuts, see below.

TABLE No. 70.

*Bolts with Whitworth Thread.**

Depth of nut = diameter of bolt = a.
Depth of lock nut = depth of bolt head = $e = 0{\cdot}7a$.
Width of head, if square
Diameter of head, if round
Width over flats, if hexagonal
} $b = 1{\cdot}4a + {\cdot}2$ inch.
Width measured over angles $c = 1{\cdot}155b$.

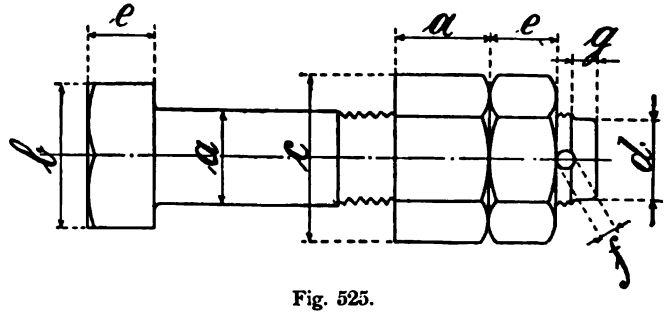


Fig. 525.

* The dimensions of nuts vary greatly in the different tables. The following table must therefore be taken as affording a basis for calculation only, and their actual dimensions may vary with the requirements of the different clients. For instance, the Imperial German Navy specifies particular sizes for nuts.

No. of threads to the inch $= n$. $\therefore$ pitch $= \frac{1}{n}$ inch.

Diameter at bottom of thread $d = a - 1{\cdot}28 \times \text{pitch} = a - \frac{1{\cdot}28}{n}$ inch.

Diameter of Bolt in inches.	Diameter at Bottom of Thread in inches.	Number of Threads per inch.	Hexagonal Nuts.		Depth of Bolt Head in inches.	Diameter of Split Pin in inches.	Depth of Point when Pin is fitted in inches.	Area at Bottom of Thread in square inches.*
			Width over Flats in inches.	Width over Angles in inches.				
a.	*d.*	*n.*	*b.*	*c.*	*e.*	*f.*	*g.*†	A.
$\frac{3}{16}$	0·134	24	·448	·5173	·164	—	—	0·014
$\frac{1}{4}$	0·186	20	·525	·6062	·2187	—	—	0·027
$\frac{5}{16}$	0·241	18	·6014	·6944	·2734	—	—	0·046
$\frac{3}{8}$	0·295	16	·7094	·8191	·3281	—	—	0·068
$\frac{7}{16}$	0·346	14	·8204	·9473	·3828	—	—	0·094
$\frac{1}{2}$	0·393	12	·9191	1·0612	·4375	—	—	0·121
$\frac{9}{16}$	0·456	12	1·011	1·1674	·4921	—	—	0·163
$\frac{5}{8}$	0·509	11	1·101	1·2713	·5468	0·157	0·236	0·203
$\frac{3}{4}$	0·622	10	1·3012	1·5024	·6562	0·177	0·275	0·304
$\frac{7}{8}$	0·733	9	1·4788	1·7075	·7656	0·177	0·275	0·422
1	0·840	8	1·6701	1·9284	·875	0·196	0·314	0·554
$1\frac{1}{8}$	0·942	7	1·8605	2·1483	·9843	0·196	0·314	0·697
$1\frac{1}{4}$	1·067	7	2·0483	2·3651	1·0937	0·236	0·354	0·894
$1\frac{3}{8}$	1·161	6	2·2146	2·5571	1·2031	0·236	0·354	1·058
$1\frac{1}{2}$	1·286	6	2·4134	2·7867	1·3125	0·275	0·433	1·299
$1\frac{5}{8}$	1·369	5	2·5763	2·9748	1·4218	0·275	0·433	1·472
$1\frac{3}{4}$	1·494	5	2·7578	3·1844	1·5312	0·315	0·472	1·753
$1\frac{7}{8}$	1·590	$4\frac{1}{2}$	3·0183	3·4852	1·6406	0·315	0·472	1·986
2	1·715	$4\frac{1}{2}$	3·1491	3·6362	1·75	0·354	0·551	2·311
$2\frac{1}{8}$	1·930	4	3·546	4·0945	1·9687	0·354	0·551	2·925
$2\frac{1}{4}$	2·180	4	3·894	4·4964	2·1875	0·393	0·630	3·732
$2\frac{3}{8}$	2·384	$3\frac{1}{2}$	4·181	4.8278	2·4062	0·393	0·630	4·464
3	2·634	$3\frac{1}{2}$	4·531	5·2319	2·625	0·472	0·708	5·450
$3\frac{1}{8}$	2·855	$3\frac{1}{4}$	4·85	5·6002	2·843	0·472	0·708	6·402
$3\frac{1}{4}$	3·105	$3\frac{1}{4}$	5·175	5·9755	3·062	0·551	0·826	7·563
$3\frac{3}{8}$	3·323	3	5·55	6·4085	3·281	0·551	0·826	8·673
4	3·573	3	5·95	6·8704	3·5	0·629	0·944	10·027
$4\frac{1}{4}$	3·804	$2\frac{7}{8}$	6·375	7·3612	3·718	0·629	0·944	11·365
$4\frac{1}{2}$	4·054	$2\frac{7}{8}$	6·825	7·8819	3·937	0·669	1·062	12·908
$4\frac{3}{4}$	4·284	$2\frac{3}{4}$	7·3	8·4293	4·156	0·669	1·062	14·404
5	4·534	$2\frac{3}{4}$	7·8	9·0066	4·375	0·708	1·181	16·146
$5\frac{1}{4}$	4·762	$2\frac{5}{8}$	8·35	9·6417	4·593	0·708	1·181	17·810
$5\frac{1}{2}$	5·012	$2\frac{5}{8}$	8·85	10·2190	4·812	0·748	1·259	19·72
$5\frac{3}{4}$	5·239	$2\frac{1}{2}$	9·45	10·9119	5·031	0·748	1·259	21·57
6	5·489	$2\frac{1}{2}$	10·0	11·5470	5·25	0·787	1·338	23·64

* Calculated from the diameter at the bottom of the thread.

† As a rule, the point starts immediately above the nut, so that there are no threads outside the nut.

TABLE No. 71.

Bolts with Fine Threads (for Connecting Rods, Crossheads, Main-bearing Bolts, &c.).

Diameter of Bolt in inches.	Diameter at Bottom of Thread in inches.	Number of Threads per inch.	Area at Bottom of Thread in square inches.	Diameter of Bolt in inches.	Diameter at Bottom of Thread in inches.	Number of Threads per inch.	Area at Bottom of Thread in square inches.
$\frac{7}{8}$	0·665	15	0·347	3	2·787	6	6·102
$\frac{7}{8}$	0·756	13	0·472	$3\frac{1}{4}$	2·995	5	7·052
1	0·901	13	0·638	$3\frac{1}{2}$	3·244	5	8·265
$1\frac{1}{8}$	1·019	12	0·816	$3\frac{3}{4}$	3·496	5	9·600
$1\frac{1}{4}$	1·118	10	0·988	4	3·716	$4\frac{1}{2}$	10·850
$1\frac{3}{8}$	1·247	10	1·223	$4\frac{1}{4}$	3·976	$4\frac{1}{2}$	12·418
$1\frac{1}{2}$	1·374	10	1·482	$4\frac{1}{2}$	4·212	$4\frac{1}{2}$	13·937
$1\frac{5}{8}$	1·484	9	1·730	$4\frac{3}{4}$	4·448	4	15·544
$1\frac{3}{4}$	1·610	9	2·036	5	4·685	4	17·239
$1\frac{7}{8}$	1·732	9	2·356	$5\frac{1}{4}$	4·921	4	19·021
2	1·818	7	2·598	$5\frac{1}{2}$	5·110	$3\frac{1}{2}$	20·573
$2\frac{1}{4}$	2·039	6	3·266	$5\frac{3}{4}$	5·400	$3\frac{1}{2}$	22·848
$2\frac{1}{4}$	2·287	6	4·109	6	5·629	$3\frac{1}{2}$	24·894
$2\frac{3}{4}$	2·539	6	5·064	7	6·653	$3\frac{1}{2}$	34·769

TABLE No. 72.

Ring Nuts.

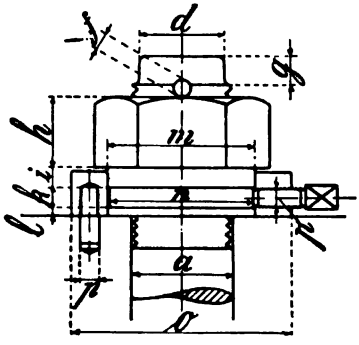


Fig. 524A.

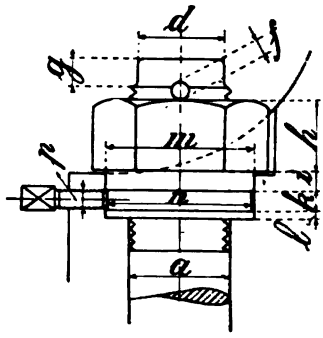


Fig. 525A.

f and *g* are the same as in Table No. 70;* *d* is the same as in Tables Nos. 70 or 71.

* As a rule the turned end begins immediately above the nut.

TABLE No. 72—*continued.*

a.	*h.*	*i.*	*k.*	*l.**	*m.*	*n.*	*o.*	*p.*
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
$\frac{3}{4}$	0·59	0·19	0·23	0·10	1·22	1·14	2·00	$\frac{1}{4}$
$\frac{7}{8}$	0·67	0·19	0·23	0·10	1·37	1·33	2·20	$\frac{1}{4}$
1	0·78	0·23	0·31	0·11	1·57	1·49	2·75	$\frac{5}{16}$
$1\frac{1}{8}$	0·86	0·23	0·31	0·11	1·69	1·61	2·87	$\frac{5}{16}$
$1\frac{1}{4}$	0·94	0·23	0·31	0·11	1·89	1·81	3·07	$\frac{5}{16}$
$1\frac{3}{8}$	1·02	0·23	0·31	0·11	2·04	1·96	3·22	$\frac{5}{16}$
$1\frac{1}{2}$	1·10	0·23	0·31	0·13	2·24	2·16	3·46	$\frac{5}{16}$
$1\frac{5}{8}$	1·18	0·31	0·35	0·13	2·44	2·36	3·85	$\frac{3}{8}$
$1\frac{3}{4}$	1·30	0·31	0·35	0·15	2·59	2·52	4·01	$\frac{3}{8}$
$1\frac{7}{8}$	1·41	0·31	0·35	0·15	2·75	2·67	4·17	$\frac{3}{8}$
2	1·49	0·31	0·35	0·19	2·91	2·83	4·33	$\frac{3}{8}$
$2\frac{1}{4}$	1·69	0·39	0·39	0·19	3·22	3·11	5·04	$\frac{1}{2}$
$2\frac{1}{2}$	1·85	0·39	0·39	0·19	3·58	3·46	5·39	$\frac{1}{2}$
$2\frac{3}{4}$	2·08	0·39	0·39	0·23	3·93	3·81	5·74	$\frac{1}{2}$
3	2·28	0·39	0·39	0·23	4·29	4·17	6·10	$\frac{1}{2}$
$3\frac{1}{4}$	2·44	0·47	0·55	0·27	4·64	4·52	6·85	$\frac{5}{8}$
$3\frac{1}{2}$	2·60	0·47	0·55	0·27	5·00	4·88	7·20	$\frac{5}{8}$
$3\frac{3}{4}$	2·75	0·47	0·55	0·27	5·31	5·19	7·52	$\frac{5}{8}$
4	2·95	0·47	0·55	0·31	5·67	5·55	7·87	$\frac{5}{8}$
$4\frac{1}{4}$	3·15	0·59	0·63	0·31	6·02	5·86	8·62	$\frac{3}{4}$
$4\frac{1}{2}$	3·34	0·59	0·63	0·31	6·37	6·22	8·97	$\frac{3}{4}$
$4\frac{3}{4}$	3·54	0·59	0·63	0·35	6·73	6·58	9·33	$\frac{3}{4}$
5	3·74	0·59	0·63	0·35	7·08	6·93	9·68	$\frac{3}{4}$
$5\frac{1}{4}$	3·93	0·63	0·78	0·39	7·44	7·28	10·43	$\frac{7}{8}$
$5\frac{1}{2}$	4·13	0·63	0·78	0·39	7·79	7·63	10·78	$\frac{7}{8}$
$5\frac{3}{4}$	4·33	0·63	0·78	0·39	8·07	7·91	11·06	$\frac{7}{8}$
6	4·52	0·63	0·78	0·47	8·42	8·26	11·41	$\frac{7}{8}$

§ 351. **Screw Spanners.**—In the equipment of the ship one or more sets of single (see Fig. 529) and double ended spanners (see Fig. 526) are provided. For large nuts, which have to be constantly taken off (connecting-rod bolts, main-bearing bolts, &c.), a light spanner is generally used for screwing them on, and a heavy spanner, which can be hammered, for tightening them up. For large heavy nuts the spanners

* In the ring nuts shown at Fig. 525A, *l* may be taken somewhat smaller.

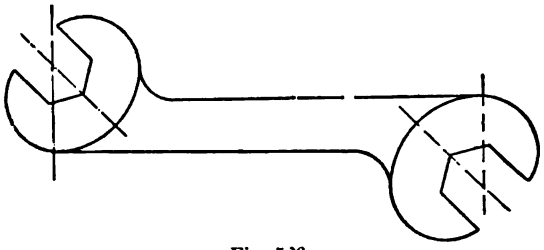


Fig. 526.

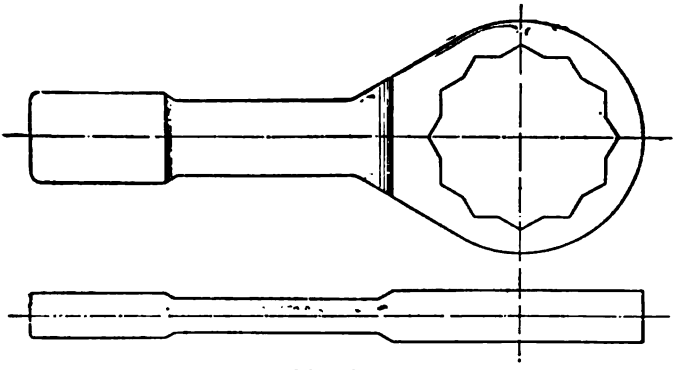


Fig. 527.

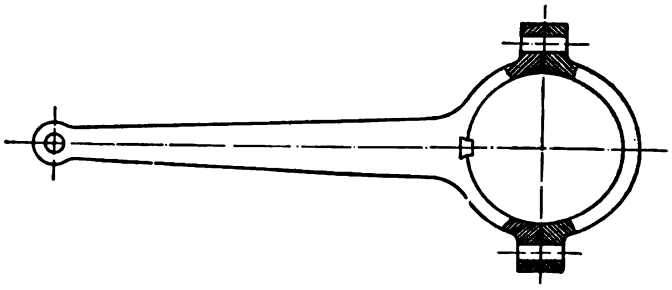


Fig. 528.

TABLE No. 73.—*Screw Spanners* (see Fig. 529).

Diameter of Bolt.	Width of Jaw. A.	B.	C.	D.	E.	F.	G.	H.	I.	K.	L.	M.	N.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
$\frac{3}{8}$	0·78	0·25	0·51	0·25	0·55	1·06	7·28	0·74	0·63	0·74	0·43	0·25	0·25
$\frac{1}{2}$	0·94	0·39	0·70	0·43	0·74	1·61	9·44	0·94	0·63	0·82	0·51	0·39	0·31
$\frac{5}{8}$	1·10	0·43	0·74	0·51	0·82	1·77	10·23	1·14	0·74	0·86	0·63	0·39	0·31
$\frac{3}{4}$	1·31	0·51	0·86	0·59	0·86	1·92	11·18	1·33	0·74	0·98	0·63	0·39	0·31
$\frac{7}{8}$	1·49	0·55	1·06	0·63	0·98	2·20	13·00	1·50	0·74	1·14	0·70	0·43	0·39
1	1·69	0·63	1·18	0·70	1·14	2·36	13·77	1·57	0·86	1·14	0·70	0·51	0·43
$1\frac{1}{8}$	1·81	0·70	1·25	0·74	1·25	2·63	15·74	1·69	0·98	1·25	0·74	0·51	0·43
$1\frac{1}{4}$	2·00	0·74	1·37	0·78	1·37	2·75	16·92	1·81	1·14	1·30	0·74	0·51	0·43
$1\frac{3}{8}$	2·16	0·74	1·49	0·82	1·61	3·22	18·11	1·92	1·25	1·37	0·86	0·55	0·51
$1\frac{1}{2}$	2·36	0·82	1·61	0·86	1·69	3·50	19·68	2·04	1·37	1·45	0·94	0·55	0·51
$1\frac{5}{8}$	2·56	0·86	1·77	0·98	1·77	3·74	21·25	2·12	1·37	1·45	0·94	0·55	0·51
$1\frac{3}{4}$	2·71	0·86	1·89	1·10	1·81	4·02	22·83	2·24	1·37	1·49	0·98	0·55	0·51
$1\frac{7}{8}$	2·91	0·94	2·00	1·18	1·92	4·25	24·01	2·36	1·49	1·49	0·98	0·55	0·51
2	3·07	0·98	2·08	1·25	2·12	4·48	25·60	2·52	1·49	1·61	1·06	0·63	0·55
$2\frac{1}{8}$	3·26	1·06	2·20	1·37	2·44	4·76	27·55	2·67	1·61	1·73	1·14	0·63	0·55
$2\frac{1}{4}$	3·42	1·14	2·36	1·49	2·56	5·11	29·52	2·83	1·77	1·85	1·18	0·63	0·55
$2\frac{3}{8}$	3·58	1·18	2·52	1·73	2·63	5·43	31·50	3·00	1·89	1·92	1·25	0·70	0·63
$2\frac{1}{2}$	3·78	1·26	2·63	1·96	2·63	5·82	33·46	3·15	2·00	2·00	1·25	0·70	0·63

are generally made in the form of ring spanners, and if there is not much room to turn them, they sometimes have twelve angles instead of six. Very large spanners (for propeller nuts, stern tubes, &c.) frequently have an eye at the end of the handle, for turning them with a rope and tackle. Such a spanner for a slotted nut is shown at Fig. 528.

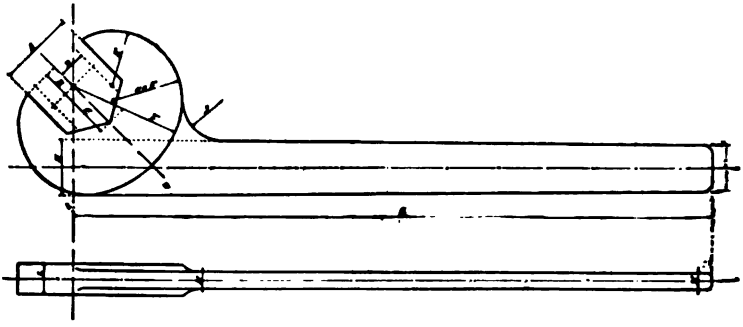


Fig. 529.

Spanners are made of steel rough ground on the grindstone and the sides in the jaws machined. In the best practice they are specially finished bright and then hardened.

Table No. 73 gives the dimensions of single-ended spanners.

§ 352. **Platforms.**—Platforms are fixed where necessary and at convenient heights. In the engine-room one is usually arranged a little above the crank shaft, at about the same level as the top of the bedplate; and in the boiler-room at about 2 feet 6 inches to 4 feet below the centre of the furnace. The platform consists of chequered plates of convenient size, so that they can easily be taken up, to get at the pipes above the double bottom. Thickness of plates $\frac{5}{32}$ to $\frac{9}{32}$ inch, and $\frac{3}{8}$ inch in the boiler-room measured over the ribs. The plates are laid on angle-iron frames, which are secured to the engine and the body of the ship, and supported where necessary on wrought-iron uprights or supports. Cocks and valves below the platform have a square end on the valve spindle, to turn them by means of a box key which is passed down through a hole in the platform, fitted with a cover.

§ 353. **Edging Plates** about $\frac{1}{8}$ inch thick and 6 to 16 inches high are put round the engine and all the openings in the platform in the engine-room. The cranks are protected by metal plates, to prevent the oil and water being splashed over the platform. These plates must be easily removable, and must obstruct the examination or feeling of the cranks as little as possible. The quick running wheels of the turning and reversing engines, &c., should be protected by metal casings.

§ 354. **The Gratings** in heavily built ships are composed of a framing of flat iron about $2\frac{1}{2}$ to 3 inches deep and $\frac{5}{16}$ to $\frac{3}{8}$ inch thick, with $\frac{3}{4}$ inch bars riveted in. Sometimes bars of wedge or T-shaped section are used. They are spaced 2 to $2\frac{1}{2}$ inches apart. Only such sections of bars should be used as are easily cleaned. In gratings where weight has to be considered, the frames may be made of ⊏-iron or light flat iron, and the bars made only $\frac{5}{8}$ inch diameter. Gratings in the boiler-room always have round iron bars about $\frac{3}{4}$ inch diameter.

Allowable width of gratings—

With round iron bars $\frac{5}{8}$ inch diameter = 18 inches.
With round iron bars $\frac{3}{4}$ inch diameter = 24 inches.

Broader gratings must be braced across the middle by flat iron bars

If wedge-shaped or T-section bars are used, the width of the gratings may be considerably increased, corresponding to the thickness of the bar used. If there is plenty of room, the gratings may be from 2 feet 6 inches to 3 feet wide, but where space is restricted they are often only 1 foot wide. The top gratings are made as light as possible, and all gratings must be provided with handrails. They are supported on wrought-iron or cast-iron brackets, spaced from 6 to 12 feet apart. In very large engines, with Stevenson's link motion, the lowest grating is a little below the valve gear, the middle one a little above the lower edge of the cylinder, and the top grating on a level with the cylinder covers. Small engines have only one grating, at the same height as the bottom of the cylinder.

§ 355. **Ladders.**—These have sides of flat iron, and rungs of cast iron, chequered plate, &c. In the boiler-room, instead of steps, there are rungs of round iron $\frac{5}{8}$ to $\frac{3}{4}$ inch diameter. Sometimes the rungs consist of one bar of round and one bar of square section side by side, the latter being in front. Ladders are placed nearly vertical in the boiler-room, or if there is sufficient space inclined at 75° to 80°. Width of the main ladder in the engine-room, 1 foot 8 inches to 2 feet 6 inches; of those in the boiler-room, 15 to 20 inches; distance between steps, 8 to 10 inches.

§ 356. **Balusters and Handrails** are provided wherever necessary. The handrails are made of smooth drawn or polished round iron or steel tubing from $\frac{7}{8}$ to 1 inch diameter. The wrought-iron balusters carrying the handrail are about 3 feet high, and have a ball head at the top, through which the handrail passes. They are secured at the bottom to the frame, or to the bracket carrying the frame, by a flange with two bolts, or by a collar, thread, and nut. The diameter of the balusters if 3 feet high, as in merchant vessels, is usually 1 inch at the top, $1\frac{1}{4}$ inch at the bottom, and diameter of ballhead about $1\frac{3}{4}$ to 2 inches. In warships the balusters are made lighter and shorter.

§ 357. **Lifting Gear over the Engines.**—In medium-sized and large merchant ships there are travelling cranes above the engines, by means of which the cylinder covers, pistons, &c., and even the upper cylinders themselves, when placed tandem one above the other, can be lifted and moved either lengthwise or athwartships. The cranes run lengthways of the ship upon I section girders placed at each side of the engine hatch. The crane girders are C or I section. The lifting machinery on the crab consists of worm gear, or of a screwed spindle only, the manœuvring of the crane being done entirely by hand. In

very large ships two cranes may be used over each engine, running on the same set of rails, and occasionally they are electrically driven.

§ 358. **Lifting Gear for Engines of Warships.**—In warships it is impossible to have a travelling crane, on account of the armoured deck. Below the latter, and immediately above the cylinder covers, there are fitted short ⊏ or I section girders placed side by side, two and two together, along the lower members of which small travellers work, with gear to lift the cylinder covers and pistons. The latter can only be taken out after they have been disconnected from the piston rods.

§ 359. **Engine Foundations.**—The stresses upon the engine foundation are produced by—

1. Weight of the engine.
2. Force exerted by the total mass and weight of the engines, due to pitching and rolling of the ship.
3. The vertical and horizontal forces and moments due to the inertia of the reciprocating and rotating parts.
4. The moment on each crank, due to the turning moment of the engine.
5. Bending moments resulting from the thrusts due to the steam pipes, in the case of two cylinders placed side by side, but not stayed. To these must be added the stresses which the foundation has to resist as part of the hull of the ship.

The stresses under 1, 2, and 5 are generally small in comparison with those under 3 and 4. These latter may, in large engines, become very considerable, especially if the oscillations of the ship are a multiple of, or synchronise with the number of revolutions of the engine.

Some idea of the extent and direction of these stresses may be obtained by plotting the moments for each position of the crank, and then constructing the polygon of forces. In order to determine the stresses transmitted to the body of the ship, the starting point may be taken at a place where the foundation of the engine merges into the double bottom of the ship. In reciprocating engines it is often sufficient to determine the greatest equivalent twisting moment of an outer crank, in relation to the plane of the adjacent inner crank. In this case the stresses upon the hull of the ship can only be small. The influence of the length of the connecting rod is here neglected, as it is not important. If an engine has strong diagonal cross braces between the different columns or entablatures, the foundation may be made much lighter, on the assumption that the longitudinal bracing converts the framing of the engine into a solid girder.

§ 360. **Construction of the Engine Foundation** (see also Bedplates).

(*a.*) *Merchant Ships with Double Bottom.*—The flanges for securing the bedplate are generally all in one horizontal plane; the bedplate being simply laid upon the plates forming the top of the double bottom. The framing of the double bottom under the engine is generally deeper than elsewhere, and a few horizontal girders, of the same depth as the double bottom, are fixed into it, over and above those required for stiffening the ship. The top plates of the double bottom are especially strengthened

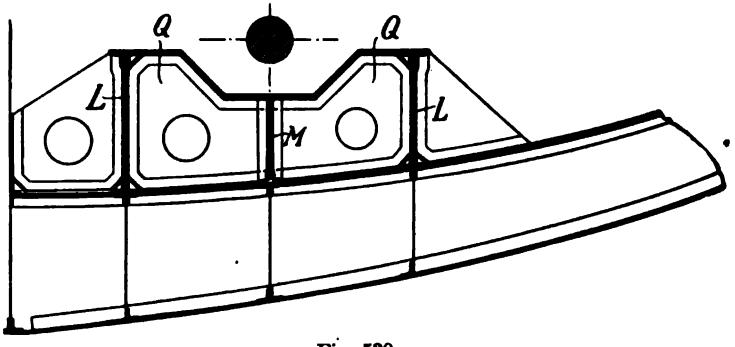


Fig. 530.

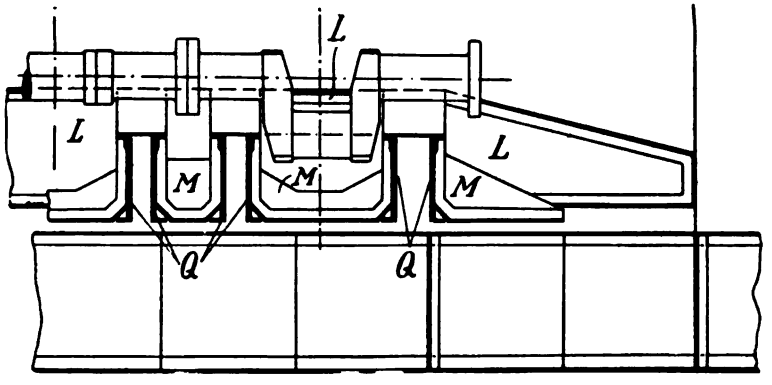


Fig. 531.

(being made up to $1\frac{1}{4}$ inch thick for fast steamers, in medium-sized merchant ships $\frac{7}{8}$ inch to 1 inch, and from $\frac{1}{2}$ inch to $\frac{5}{8}$ inch in smaller ships), in order to accommodate the holding-down bolts, &c. (see Bedplates).

(*b.*) *Warships with Double Bottom.*—The framing of the engine seating is, as a rule, deeper in the middle than at the sides, and is braced fore and aft by longitudinal beams. These longitudinal beams are

situated immediately under the longitudinal members of the bedplate, and rest on, and are riveted to the double bottom. Under the bedplate these longitudinal girders are usually braced by cross box girders, which are cut away to accommodate the engine seating. The space between the girders and the cross frame of the engine bedplate is fitted with wooden packing pieces (compare Figs. 530, 531).

(*c.*) If there is *no double bottom*, the longitudinal girders are laid across the frames, and strongly connected to them (there should be at least four rivets at each intersection). It is best to fix the longitudinals to the frames or the body of the ship by intercostals reaching down between the frames. If the engines are heavy, and the bedplate is level

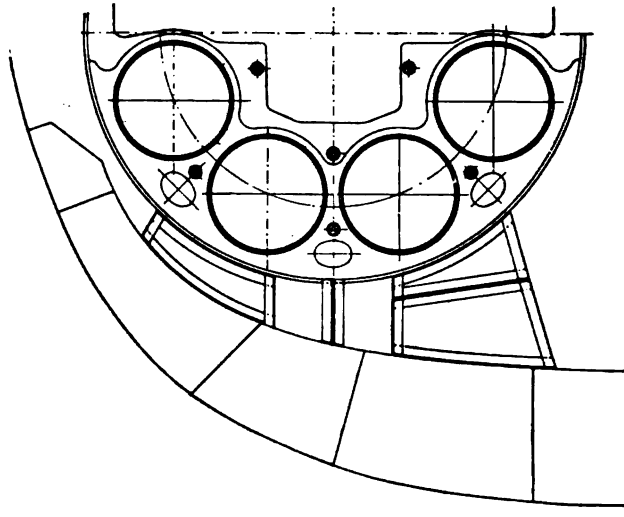


Fig. 532.

along the bottom, a series of longitudinal girders may be placed above and across the frames, and plated over, so as to form a surface for the bedplate to rest on.

§ 361. **Boiler Seatings** (see Fig. 532).—The bearers, composed of angle irons and plates, extend over about one-quarter of the circumference, and should be so arranged that they do not cover the riveted seams, stay bolts, &c. The number of bearers, composing the seatings on either side of the boiler, depends on the size, and especially on the weight of the boiler, and may vary from two to six. The size of the angle iron used is from $2\frac{3}{4} \times 2\frac{3}{4} \times \frac{3}{8}$ in. to $4 \times 4 \times \frac{1}{2}$ in.; thickness of the iron plates from $\frac{5}{16}$ to $\frac{1}{2}$ in. The seating nearest the centre of the ship

is generally made in two parts, that the boiler may be more easily stowed in the wing of the ship.

Each boiler bearer should, as far as possible, rest on a frame (or on projecting parts of the ship's bottom); if this cannot be done, they should be supported on girders carried by the frames, so as to transmit the strains directly to the latter. Above the double bottom the bearers should be connected by side plates, either riveted or bolted on. In order that the boiler may be got at between the bearers, holes are cut in the connecting plates and bearers. To make the lower part of the boiler shell accessible, the lower edge of the boiler should, if possible, be at least 12 inches above the double bottom of the ship.

To keep the boiler in place when the ship rolls, &c., from two to four eyes are riveted to the upper part of the boiler shell, and are secured with tie bars to the nearest deck beams, "stringers," or to the adjacent boilers. These eyes may also be used for lifting the boiler. Another method of securing the boiler is to rivet four eyes to it immediately above the bearers, and connect them with the latter by means of short ties. In this way the boiler is fastened independently of the deck, which always "gives" more or less, and also independently of the skin of the ship. To prevent the boiler shifting longitudinally, strong angle-iron brackets are riveted or bolted to the double bottom of the ship, and butt against the front and back of the boiler.

§ 362. **Lubrication* of the Steam Spaces.**—This is generally effected with mineral oil, to which graphite is sometimes added. The oil is forced into the lubricating pipes by a piston working in a cylinder, and driven by steam or by means of a special lever from one of the working parts of the engine. The pipe leads either to the main stop valve *only* (to lubricate the steam), or in larger engines to each valve chest as well as the main stop valve. The cylinders are either connected to the oil pipe referred to above, or they have separate lubricators so designed that oil finds its way into the cylinders, in spite of the pressure of steam existing in them. The lubricator generally has an arrangement whereby oil can be quickly forced out by hand or by steam pressure, should a piston or valve require special lubrication, and there should also be some means of shutting off the pipe supplying any particular part of the engine. To prevent steam from say the H.P. or the M.P. cylinders reaching the L.P. through the branches of the main oil pipe, there is a small non-return valve at the mouth of each branch pipe, which is forced open by the pressure in the lubricator.

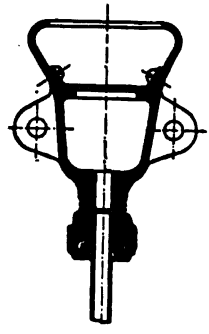


Fig. 533.

The piston rods are swabbed from time to time with mineral oil, which is retained for a short time by the wicks attached to the outer end of the stuffing boxes.

§ 363. **Lubrication of other Parts.**—Almost all the other parts of the engine are lubricated with *vegetable* or *animal* oil, which is applied either by *hand*, by *siphon lubricators*, or by *automatic sight-feed lubricators*. For the rotating or non-reciprocating parts, the oil flows from the oil boxes direct to the bearings. The reciprocating parts are supplied with oil from a stationary lubricator. A pipe carries the oil from the latter to an open cup fixed to the reciprocating part, or the oil is wiped off a small brush or feather at the end of the pipe into the cup. Several small oil pipes are usually run from one large lubricator, which should be easy of access. In reciprocating parts (such as the crosshead, &c.) some arrangement should be provided, to prevent the oil being flung

* Compare the arrangements of engines and photographs in former chapters.

out of the lubricator. A small piece of perforated metal or wire gauze is generally placed in the upper part of the cup, to prevent any dirt finding its way into the bearings (see Fig. 533).

The oil cups on the reciprocating parts must be solidly screwed in, so that they may not work loose, and fly off when the engine is running; the oil pipes must also be secured in a substantial manner. The most important parts of the engine, such as the crank and crank-shaft bearings, are fitted with arrangements for oiling by hand, besides the siphon lubricators. Care must be taken that the oil pipes reach into the brasses, and fit them tightly.

The oil is distributed throughout the bearings by means of grooves, which are cut in the working surface of the bearing (the grooves should be from $\frac{1}{4}$ to $\frac{3}{4}$ inch wide, according to the size of the bearing, and half as deep). The bearings must be so arranged that the oil cannot find its way out sideways, but is forced to distribute itself evenly over the bearing.

Internal Diameter of Lubricating Pipes.

	Crank and Main Bearings.	Valve Gear, &c.
	Inch.	Inch.
Engines below 1,000 I.H.P.	$\frac{3}{16}$ to $\frac{3}{8}$	$\frac{3}{16}$ to $\frac{5}{16}$
„ from 1,000 to 3,000 I.H.P.	$\frac{5}{16}$ to $\frac{9}{16}$	$\frac{1}{4}$ to $\frac{1}{2}$
„ „ 3,000 to 7,000 „	$\frac{1}{2}$ to $\frac{11}{16}$	$\frac{3}{8}$ to $\frac{5}{8}$
„ „ 7,000 I.H.P. and upwards	$\frac{5}{8}$ to $\frac{3}{4}$	$\frac{1}{2}$ to $\frac{5}{8}$

The Plummer blocks are sometimes lubricated with solid grease, as well as with oil.

Bearings subjected to *light strains* are often only provided with small holes fitted with Stauffer lubricators.

Worm gear should, if possible, be run in an oil bath.

Besides the arrangements here described, a large number of special types of lubricators are also employed.

§ 364. **Ash Hoists.**—In *smaller ships* the ashes are drawn up to the deck in buckets with chains by small hand winches, through one of the ventilators, and then thrown overboard. In *larger ships* the winches are worked by steam (diameter of cylinder and stroke about 4 inches). Number:—Either one for each stokehold or one for every two stokeholds, if conveniently situated near each other. The winches are fixed in the boiler-room casings, at the level of the upper deck. The buckets are hoisted up in the ventilators by means of wire ropes. The winches are so arranged that the steam is cut off automatically, when the bucket reaches the top.

§ 365. **Ash-ejectors.**—In recent large ships the ashes are discharged by what are known as "ash-ejectors" (see Fig. 534). They are shovelled into a cast-iron hopper T, on the top of which is a strong cast-iron grating, through which the ashes fall into the hopper, and on which they can be further broken, if necessary. At the bottom of the hopper T is a jet D, through which, in the direction of the discharge pipe R, a stream of water at a pressure of 150 to 200 lb. per square inch is forced, and carries the ashes with it through the pipe discharging them overboard at least 3 feet above the water line. The pipe can be closed by a flap valve K, so that water cannot find its way to the stokehold in bad weather. The upper part of the pipe in the bend is formed of a separate piece S, which can be easily renewed when worn out. The water under pressure is delivered by the auxiliary feed, or other special pump.

To prevent water finding its way from the ash-ejector into the stokehold, the water pressure must be turned on suddenly. This may be effected by means of an ordinary cock, a valve, or an adjustable differential piston.*

In the latter arrangement (see Fig. 535) the water under pressure enters from the pump at A. The spaces G and H are placed in communication through openings E and F in the hollow piston. In the position shown in the drawing, the water passes through E and F to pipe C, which

* Patentee Howaldtswerke, Kiel.

leads overboard through a cock and non-return valve. If the regulating cock *a* be turned through 90° so that passages *p* and *r* are made to communicate, the water under pressure drives the piston, with diameter D, to the right, and closes pipe C. Simultaneously pipe B is opened, and

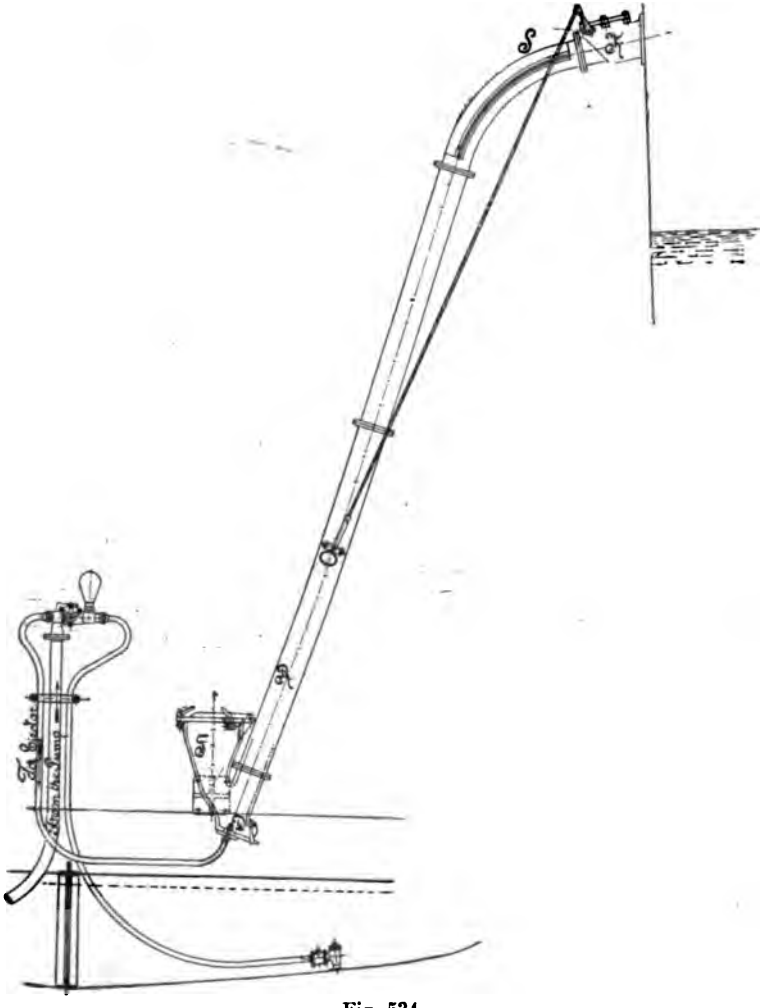


Fig. 534.

the water led off to the jet in the ash-ejector. Diameter of nozzle of jet $\frac{1}{2}$ to $\frac{3}{4}$ inch. Diameter of discharge pipe, 4 to 6 inches. Thickness about $\frac{5}{16}$ to 1 inch. Inclination of the discharge pipe to the horizontal, about 60°. As the bent upper part of this pipe requires frequent

renewal, it must be easily accessible. Diameter of hopper at the top, about 14 to 18 inches. Diameter of delivery pipe from the pump, about 2 to 2½ inches. Number of ash-ejectors, the same as of steam ash hoists.

§ 366. **Ventilation of the Engine and Boiler Rooms.**—The engine and boiler rooms of merchant vessels are almost always ventilated by the natural draught produced by the motion of the ship, or

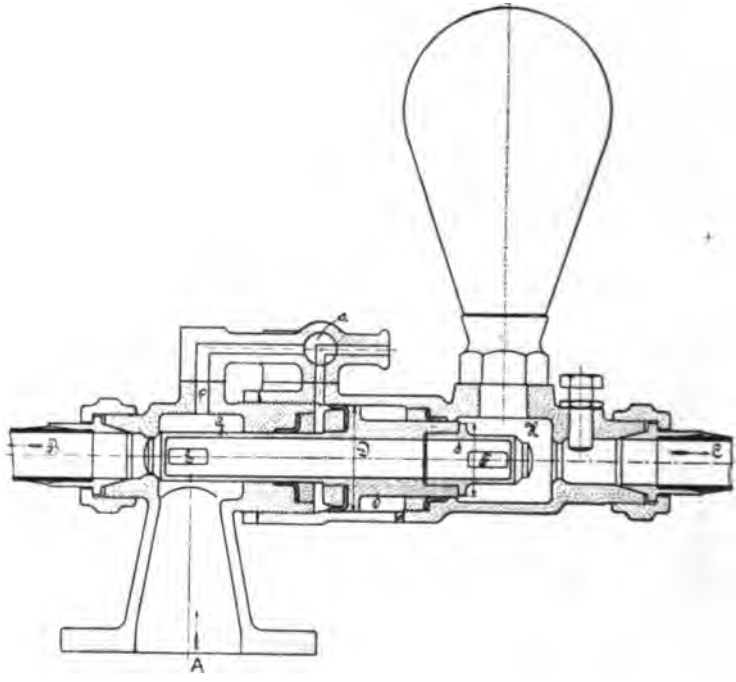


Fig. 535.

by the force of the wind. The air is led below by means of thin iron pipes of large diameter, fitted at their upper ends with revolving cowls.

With a velocity of air of about 10 feet per second, *the cross section of the ventilators* should be sufficient, to deliver into the stokehold about 350 cubic feet of air per pound of coal burnt. Besides the engine-room hatch, which has to be closed in bad weather, special ventilators with cowls are fitted.

§ 366A. **Area of Engine-room Ventilators.**—If G be the floor

TABLE No. 74.—*Fans in Boiler-rooms.*

Type of Ship.	Gunboat, building in 1903.	Small Cruiser, building in 1903.	Large Cruiser, building in 1903.		Battleship, building in 1903.	
Main Engine max. I.H.P. - -	2 × 650	2 × 5,000	3 × 6,333		2 × 5,333	
Number of fans - - -	1	6*	After engine-room 2	2 forward engine-rooms 2 × 2	After engine room 2	2 forward engine-rooms 2 × 2
External diameter of fan - -	Inches. 43·3	Inches. 39·3	Inches. 63	Inches. 59	Inches. 70	Inches. 55
Internal diameter of fan - -	2 × 23·6	2 × 22	2 × 34·6	2 × 34·6	2 × 37·4	2 × 29·5
Diameter of H.P. cylinder - -	3·5	3·5	5	5·5	4·3	3·5
Diameter of L.P. cylinder - -	—	—	8·2	—	—	—
Stroke - - - - - -	3·5	3·5	4·7	4·7	4·3	4·3

* Two in starboard room, two in port room, two in auxiliary engine-room.

space of the engine-room, A the total area of all the ventilators leading into it, then—

$$A = \frac{G}{160} \text{ to } \frac{G}{80}, \text{ generally} = \frac{G}{120} \text{ to } \frac{G}{100}$$

These values naturally vary considerably, according to the different sizes of the engine-room. In the case of very small engines, the skylight alone is often sufficient for ventilating purposes.

§ 367. **Ventilation of the Engine and Boiler Rooms in Warships.**—In warships, if the boilers are worked with the closed stokehold system, the boiler-rooms are ventilated by the fans delivering air under pressure into the stokehold (compare page 567, Closed Stokehold System). The engine-rooms of warships are also ventilated by special fans, which deliver air, where specially required, by means of ventilators made of thin plating. Table No. 74, compiled from the latest arrangements used in the German Imperial Navy, gives details of the dimensions of these fans.

§ 368. German Lloyd's Rules for Spare Gear for Engines and Boilers.

Steamers registered at the German Lloyd's must have on board the following spare gear for engines and boilers :—

(*a.*) *Steamships classed* L *and* Atl. (*Ocean-going and Atlantic Steamers*).

1 set of crank-pin brasses.
1 set of crosshead pin brasses.
1 set of bolts and nuts for crank-pin end of connecting rod.
1 „ „ crosshead end „
1 set of main-bearing bolts and nuts.
1 set of bolts for each size of couplings.
1 set of springs and rings for each piston.
1 valve rod for each cylinder. If all the valve rods are of the same size, one spare rod is sufficient.
1 air-pump piston rod or connecting rod.
1 circulating-pump piston rod or connecting rod.
1 set of valves for air pump.
1 set of valves for circulating pump. If there is a centrifugal instead of a plunger circulating pump, brasses and bolts for the latter of every size, as well as a valve rod, piston, and piston rod.
1 set of feed-pump valves. If the pumps are of the same size, one set to every two pumps.
1 set of bilge-pump valves. If the pumps are of the same size, one set to every two pumps.
1 set of feed-check valves.
1 complete set of straps to each pump and lever.
1 safety-valve spring if all the safety valves are of the same size; if not, one spring to each valve.
2 % of condenser tubes and ferrules.
2 % of boiler tubes.
6 sets of water-gauge glasses with packing rings, for each boiler.
½ set of fire bars.
10 % each of bolts and studs for the air and circulating pumps with their nuts; half a bundle of hoop iron; one ¼ inch thick and one

$\frac{1}{8}$ inch thick iron plates; 2 bars of flat and 3 of round iron; half bar of steel; one dozen each of screws, nuts, and washers of $\frac{1}{2}$ inch, $\frac{5}{8}$ inch, $\frac{7}{8}$ inch, and 1 inch diameter respectively.

To every 50 boiler tubes, including stay tubes, one tube stopper.

Of engineer's tools, sufficient to execute small repairs on board.

It is also recommended to have in reserve—1 crank shaft; 1 tail shaft; 1 propeller or half set of blades, if the latter are not cast in one with the boss.

(*b.*) *Steamships Class* K (*Large Coasting Steamers*).

1 set of crank-pin bolts and nuts.

1 „ crosshead „

1 „ main-bearing „

1 valve rod for each cylinder. If all the valve rods are of the same size, one is sufficient.

1 set of air-pump valves.

1 „ circulating-pump valves.

1 „ feed-pump valves.

If the pumps are all of the same size, one set of valves to every two pumps.

Piston bolts, set screws, bolts, and nuts for the air and circulating pumps; 10 °/₀ of each of the number in use.

2 °/₀ of the number of condenser tubes, with packing for the tube ends.

4 sets of water-gauge glasses with packing rings, for each boiler.

½ set of fire bars.

1 boiler-tube stopper to every 50 boiler tubes.

A stock of flat and round iron bars, &c., and the necessary engineer's tools.

(*c.*) *Steamships Class k* (*Small Coasting Steamers*).

Bolts for every size of coupling.

1 crank-pin bolt.

1 crosshead bolt.

1 main-bearing bolt.

Half set of circulating and air pump valves, of bilge and feed pump valves, to each pump, or if all the pumps are of the same size, one set to every two pumps.

2 sets of water-gauge glasses with packing rings, for each boiler.

½ set of fire bars.

2 boiler-tube stoppers.

§ 369. Lloyd's Rules for Spare Gear.

Spare gear is not required in steam yachts. The spare gear mentioned in the following list will be required to be carried in all steam vessels classed in the Society's Register Book, viz.:—

2 connecting-rod or piston-rod top end bolts and nuts.
2 „ „ bottom end bolts and nuts.
2 main-bearing bolts.
1 set of coupling bolts.
1 set of feed and bilge pump valves.
1 set of piston springs (where common springs are used).
A quantity of assorted bolts and nuts.
Iron of various sizes.

In addition to the foregoing, it is recommended that the following articles be carried, with a view to expedite repairs and lessen delay in distant ports, viz.:—

Crank shaft.
Propeller shaft.
Propeller, or full set of blades.
Stern bush or lignum vitæ lining for bush.
1 pair of connecting-rod brasses.
1 pair of crosshead brasses.
1 set of link brasses.
1 eccentric strap complete.
Air pump rod.
Circulating pump rod.
H.P. valve spindle.
L.P. valve spindle.
1 set of check valves.
6 cylinder cover bolts.
6 junk ring bolts.
4 valve chest cover bolts.
2 dozen condenser tubes.
1 cylinder escape valve and spring.
1 set of safety valve springs.

PART VIII.

VARIOUS TABLES.

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N.B.—The missing Tables refer to data which it has not been thought necessary to include in the English Edition.

As the German plates have been used for Tables I. to XXI., the decimal points are indicated by commas.

Table No. I.—Squares, Cubes, Square Roots, Cube Roots, Reciprocals, Natural Logarithms, Circumferences, Areas of Circles from 1 to 1000.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
0	0	0	0,0000	0,0000	∞	$-\infty$	0,0	0,000	0,0000	0
1	1	1	1,0000	1,0000	1,00000	0,0000	0,1	0,314	0,0079	1
2	4	8	1,4142	1,2599	0,50000	0,6931	0,2	0,628	0,0314	2
3	9	27	1,7321	1,4422	0,33333	1,0986	0,3	0,942	0,0707	3
4	16	64	2,0000	1,5874	0,25000	1,3863	0,4	1,257	0,1257	4
5	25	125	2,2361	1,7100	0,20000	1,6094	0,5	1,571	0,1964	5
6	36	216	2,4495	1,8171	0,16667	1,7918	0,6	1,885	0,2827	6
7	49	343	2,6458	1,9129	0,14286	1,9459	0,7	2,199	0,3848	7
8	64	512	2,8284	2,0000	0,12500	2,0794	0,8	2,513	0,5026	8
9	81	729	3,0000	2,0801	0,11111	2,1972	0,9	2,827	0,6362	9
10	100	1 000	3,1623	2,1544	0,10000	2,3026	1,0	3,142	0,7854	10
11	121	1 331	3,3166	2,2240	0,09091	2,3979	1,1	3,456	0,9503	11
12	144	1 728	3,4641	2,2894	0,08333	2,4849	1,2	3,770	1,1310	12
13	169	2 197	3,6056	2,3513	0,07692	2,5649	1,3	4,084	1,3278	13
14	196	2 744	3,7417	2,4101	0,07143	2,6391	1,4	4,398	1,5394	14
15	225	3 375	3,8730	2,4662	0,06667	2,7081	1,5	4,712	1,7671	15
16	256	4 096	4,0000	2,5198	0,06250	2,7726	1,6	5,027	2,0106	16
17	289	4 913	4,1231	2,5713	0,05882	2,8332	1,7	5,341	2,2698	17
18	324	5 832	4,2426	2,6207	0,05556	2,8904	1,8	5,655	2,5447	18
19	361	6 859	4,3589	2,6684	0,05263	2,9444	1,9	5,969	2,8353	19
20	400	8 000	4,4721	2,7144	0,05000	2,9957	2,0	6,283	3,1416	20
21	441	9 261	4,5826	2,7589	0,04762	3,0445	2,1	6,597	3,4636	21
22	484	10 648	4,6904	2,8020	0,04545	3,0910	2,2	6,912	3,8013	22

23	529	12 167	4,7958	2,8439	0,01348	3,1355	2,3	7,226	4,1548	23
24	576	13 824	4,8990	2,8845	0,04167	3,1781	2,4	7,540	4,5239	24
25	625	15 625	5,0000	2,9240	0,01000	3,2189	2,5	7,854	4,9087	25
26	676	17 576	5,0990	2,9625	0,03846	3,2581	2,6	8,168	5,3093	26
27	729	19 683	5,1962	3,0000	0,03704	3,2958	2,7	8,482	5,7256	27
28	784	21 952	5,2915	3,0366	0,03571	3,3322	2,8	8,796	6,1575	28
29	841	24 389	5,3852	3,0723	0,03448	3,3673	2,9	9,111	6,6052	29
30	900	27 000	5,4772	3,1072	0,03333	3,4012	3,0	9,425	7,0686	30
31	961	29 791	5,5678	3,1414	0,03226	3,4340	3,1	9,739	7,5477	31
32	1 024	32 768	5,6569	3,1748	0,03125	3,4657	3,2	10,05	8,0425	32
33	1 089	35 937	5,7446	3,2075	0,03030	3,4965	3,3	10,37	8,5530	33
34	1 156	39 304	5,8310	3,2396	0,02941	3,5264	3,4	10,68	9,0792	34
35	1 225	42 875	5,9161	3,2711	0,02857	3,5553	3,5	11,00	9,6211	35
36	1 296	46 656	6,0000	3,3019	0,02778	3,5835	3,6	11,31	10,1790	36
37	1 369	50 653	6,0828	3,3322	0,02703	3,6109	3,7	11,62	10,752	37
38	1 444	54 872	6,1644	3,3620	0,02632	3,6376	3,8	11,94	11,341	38
39	1 521	59 319	6,2450	3,3912	0,02564	3,6636	3,9	12,25	11,946	39
40	1 600	64 000	6,3246	3,4200	0,02500	3,6889	4,0	12,57	12,566	40
41	1 681	68 921	6,4031	3,4482	0,02439	3,7136	4,1	12,88	13,203	41
42	1 764	74 088	6,4807	3,4760	0,02381	3,7377	4,2	13,19	13,854	42
43	1 849	79 507	6,5574	3,5034	0,02326	3,7612	4,3	13,51	14,522	43
44	1 936	85 184	6,6332	3,5303	0,02273	3,7842	4,4	13,82	15,205	44
45	2 025	91 125	6,7082	3,5569	0,02222	3,8067	4,5	14,14	15,904	45
46	2 116	97 336	6,7823	3,5830	0,02174	3,8286	4,6	14,45	16,619	46
47	2 209	103 823	6,8557	3,6088	0,02128	3,8501	4,7	14,77	17,349	47
48	2 304	110 592	6,9282	3,6342	0,02083	3,8712	4,8	15,08	18,096	48
49	2 401	117 649	7,0000	3,6593	0,02041	3,8918	4,9	15,39	18,857	49
50	2 500	125 000	7,0711	3,6840	0,02000	3,9120	5,0	15,71	19,635	50

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\left.\begin{matrix}\log\\ \text{nat}\end{matrix}\right\}n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
51	2 601	132 651	7,1414	3,7084	0,01961	3,9318	5,1	16,02	20,428	51
52	2 704	140 608	7,2111	3,7325	0,01923	3,9512	5,2	16,34	21,237	52
53	2 809	148 877	7,2801	3,7563	0,01887	3,9703	5,3	16,65	22,062	53
54	2 916	157 464	7,3485	3,7798	0,01852	3,9890	5,4	16,96	22,902	54
55	3 025	166 375	7,4162	3,8030	0,01818	4,0073	5,5	17,28	23,758	55
56	3 136	175 616	7,4833	3,8259	0,01786	4,0254	5,6	17,59	24,630	56
57	3 249	185 193	7,5498	3,8485	0,01754	4,0431	5,7	17,91	25,518	57
58	3 364	195 112	7,6158	3,8709	0,01724	4,0604	5,8	18,22	26,421	58
59	3 481	205 379	7,6811	3,8930	0,01695	4,0775	5,9	18,54	27,340	59
60	3 600	216 000	7,7460	3 9149	0,01667	4,0943	6,0	18,85	28,274	**60**
61	3 721	226 981	7,8102	3,9365	0,01639	4,1109	6,1	19,16	29,225	61
62	3 844	238 328	7,8740	3,9579	0,01613	4,1271	6,2	19,48	30,191	62
63	3 969	250 047	7,9373	3,9791	0,01587	4,1431	6,3	19,79	31,172	63
64	4 096	262 144	8,0000	4,0000	0,01563	4,1589	6,4	20,11	32,170	64
65	4 225	274 625	8,0623	4,0207	0,01538	4,1744	6,5	20,42	33,183	65
66	4 356	287 496	8,1240	4,0412	0,01515	4,1897	6,6	20,73	34,212	66
67	4 489	300 763	8,1854	4,0615	0,01493	4,2047	6,7	21,05	35,257	67
68	4 624	314 432	8,2462	4,0817	0,01471	4,2195	6,8	21,36	36,317	68
69	4 761	328 509	8,3066	4,1016	0,01449	4,2341	6,9	21,68	37,393	69
70	4 900	343 000	8,3666	4,1218	0,01429	4,2485	7,0	21,99	38,485	**70**
71	5 041	357 911	8,4261	4,1408	0,01408	4,2627	7,1	22,31	39,592	71
72	5 184	373 248	8,4853	4,1602	0,01389	4,2767	7,2	22,62	40,715	72
73	5 329	389 017	8,5440	4,1793	0,01370	4,2905	7,3	22,93	41,854	73

74	5 476	405 224	8,6023	4,1983	0,01351	4,8041	7,4	23,25	43,008	74
75	5 625	421 875	8,6603	4,2172	0,01333	4,8175	7,5	23,56	44,179	75
76	5 776	438 976	8,7178	4,2358	0,01316	4,8307	7,6	23,88	45,365	76
77	5 929	456 533	8,7750	4,2543	0,01299	4,3438	7,7	24,19	46,566	77
78	6 084	474 552	8,8318	4,2727	0,01282	4,3567	7,8	24,50	47,784	78
79	6 241	493 039	8,8882	4,2908	0,01266	4,3694	7,9	24,82	49,017	79
80	6 400	512 000	8,9443	4,3089	0,01250	4,3820	8,0	25,13	50,265	**80**
81	6 561	531 441	9,0000	4,3267	0,01235	4,3944	8,1	25,45	51,530	81
82	6 724	551 368	9,0554	4,3445	0,01220	4,4067	8,2	25,76	52,810	82
83	6 889	571 787	9,1104	4,3621	0,01205	4,4188	8,3	26,08	54,106	83
84	7 056	592 704	9,1652	4,3795	0,01190	4,4308	8,4	26,39	55,418	84
85	7 225	614 125	9,2195	4,3968	0,01176	4,4427	8,5	26,70	56,745	85
86	7 396	636 056	9,2736	4,4140	0,01163	4,4543	8,6	27,02	58,088	86
87	7 569	658 503	9,3274	4,4310	0,01149	4,4659	8,7	27,33	59,447	87
88	7 744	681 472	9,3808	4,4480	0,01136	4,4773	8,8	27,65	60,821	88
89	7 921	704 969	9,4340	4,4647	0,01124	4,4886	8,9	27,96	62,211	89
90	8 100	729 000	9,4868	4,4814	0,01111	4,4998	9,0	28,27	63,617	**90**
91	8 281	758 571	9,5394	4,4979	0,01099	4,5109	9,1	28,59	65,039	91
92	8 464	778 688	9,5917	4,5144	0,01087	4,5218	9,2	28,90	66,476	92
93	8 649	804 357	9,6437	4,5307	0,01075	4,5326	9,3	29,22	67,929	93
94	8 836	830 584	9,6954	4,5468	0,01064	4,5433	9,4	29,53	69,398	94
95	9 025	857 375	9,7468	4,5629	0,01053	4,5539	9,5	29,85	70,882	95
96	9 216	884 736	9,7980	4,5789	0,01042	4,5643	9,6	30,16	72,382	96
97	9 409	912 673	9,8489	4,5947	0,01031	4,5747	9,7	30,47	73,898	97
98	9 604	941 192	9,8995	4,6104	0,01020	4,5850	9,8	30,79	75,430	98
99	9 801	970 299	9,9499	4,6261	0,01010	4,5951	9,9	31,10	76,977	99
100	10 000	1 000 000	10,0000	4,6416	0,01000	4,6052	10,0	31,42	78,540	**100**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
101	10 201	1 030 301	10,0499	4,6570	0,00990	4,6151	10,1	31,73	80,118	101
102	10 404	1 061 208	10,0995	4,6723	0,00980	4,6250	10,2	32,04	81,713	102
103	10 609	1 092 727	10,1489	4,6875	0,00971	4,6347	10,3	32,36	83,323	103
104	10 816	1 124 864	10,1980	4,7027	0,00962	4,6444	10,4	32,67	84,949	104
105	11 025	1 157 625	10,2470	4,7177	0,00952	4,6540	10,5	32,99	86,590	105
106	11 236	1 191 016	10,2956	4,7326	0,00943	4,6634	10,6	33,30	88,247	106
107	11 449	1 225 043	10,3441	4,7475	0,00935	4,6728	10,7	33,62	89,920	107
108	11 664	1 259 712	10,3923	4,7622	0,00926	4,6821	10,8	33,93	91,609	108
109	11 881	1 295 029	10,4403	4,7769	0,00917	4,6913	10,9	34,24	93,313	109
110	**12 100**	**1 331 000**	10,4881	4,7914	0,00909	4,7005	11,0	34,56	95,033	**110**
111	12 321	1 367 631	10,5357	4,8059	0,00901	4,7095	11,1	34,87	96,769	111
112	12 544	1 404 928	10,5830	4,8203	0,00893	4,7185	11,2	35,19	98,520	112
113	12 769	1 442 897	10,6301	4,8346	0,00885	4,7274	11,3	35,50	100,287	113
114	12 996	1 481 544	10,6771	4,8488	0,00877	4,7362	11,4	35,81	102,070	114
115	13 225	1 520 875	10,7238	4,8629	0,00870	4,7449	11,5	36,13	103,869	115
116	13 456	1 560 896	10,7703	4,8770	0,00862	4,7536	11,6	36,44	105,683	116
117	13 689	1 601 613	10,8167	4,8910	0,00855	4,7622	11,7	36,76	107,513	117
118	13 924	1 643 032	10,8628	4,9049	0,00847	4,7707	11,8	37,07	109,359	118
119	14 161	1 685 159	10,9087	4,9187	0,00840	4,7791	11,9	37,38	111,220	119
120	**14 400**	**1 728 000**	10,9545	4,9324	0,00833	4,7875	12,0	37,70	113,097	**120**
121	14 641	1 771 561	11,0000	4,9461	0,00826	4,7958	12,1	38,01	114,990	121
122	14 884	1 815 848	11,0454	4,9597	0,00820	4,8040	12,2	38,33	116,899	122
123	15 129	1 860 867	11,0905	4,9732	0,00813	4,8122	12,3	38,64	118,823	123

124	15 376	1 906 624	11,1355	4,9866	0,00806	4,8203	12,4	38,96	120,763	124
125	15 625	1 953 125	11,1803	5,0000	0,00800	4,8283	12,5	39,27	122,72	125
126	15 876	2 000 376	11,2250	5,0133	0,00794	4,8363	12,6	39,58	124,69	126
127	16 129	2 048 383	11,2694	5,0265	0,00787	4,8442	12,7	39,90	126,68	127
128	16 384	2 097 152	11,3137	5,0397	0,00781	4,8520	12,8	40,21	128,68	128
129	16 641	2 146 689	11,3578	5,0528	0,00775	4,8598	12,9	40,53	130,70	129
130	16 900	2 197 000	11,4018	5,0658	0,00769	4,8675	13,0	40,84	132,73	**130**
131	17 161	2 248 091	11,4455	5,0788	0,00763	4,8752	13,1	41,15	134,78	131
132	17 424	2 299 968	11,4891	5,0916	0,00758	4,8828	13,2	41,47	136,85	132
133	17 689	2 352 637	11,5326	5,1045	0,00752	4,8903	13,3	41,78	138,93	133
134	17 956	2 406 104	11,5758	5,1172	0,00746	4,8978	13,4	42,10	141,03	134
135	18 225	2 460 375	11,6190	5,1299	0,00741	4,9053	13,5	42,41	143,14	135
136	18 496	2 515 456	11,6619	5,1426	0,00735	4,9127	13,6	42,73	145,27	136
137	18 769	2 571 353	11,7047	5,1551	0,00730	4,9200	13,7	43,04	147,41	137
138	19 044	2 628 072	11,7473	5,1676	0,00725	4,9273	13,8	43,35	149,57	138
139	19 321	2 685 619	11,7898	5,1801	0,00719	4,9345	13,9	43,67	151,75	139
140	19 600	2 744 000	11,8322	5,1925	0,00714	4,9416	14,0	43,98	153,94	**140**
141	19 881	2 803 221	11,8743	5,2048	0,00709	4,9488	14,1	44,30	156,15	141
142	20 164	2 863 288	11,9164	5,2171	0,00704	4,9558	14,2	44,61	158,37	142
143	20 449	2 924 207	11,9583	5,2293	0,00699	4,9628	14,3	44,92	160,61	143
144	20 736	2 985 984	12,0000	5,2415	0,00694	4,9698	14,4	45,24	162,86	144
145	21 025	3 048 625	12,0416	5,2536	0,00690	4,9767	14,5	45,55	165,13	145
146	21 316	3 112 136	12,0830	5,2656	0,00685	4,9836	14,6	45,87	167,42	146
147	21 609	3 176 523	12,1244	5,2776	0,00680	4,9904	14,7	46,18	169,72	147
148	21 904	3 241 792	12,1655	5,2896	0,00676	4,9972	14,8	46,50	172,03	148
149	22 201	3 307 949	12,2066	5,3015	0,00671	5,0039	14,9	46,81	174,37	149
150	22 500	3 375 000	12,2474	5,3133	0,00667	5,0106	15,0	47,12	176,71	**150**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
151	22 801	3 442 951	12,2882	5,3251	0,00662	5,0173	15,1	47,44	179,08	151
152	23 104	3 511 808	12,3288	5,3368	0,00658	5,0239	15,2	47,75	181,46	152
153	23 409	3 581 577	12,3693	5,3485	0,00654	5,0304	15,3	48,07	183,85	153
154	23 716	3 652 264	12,4097	5,3601	0,00649	5,0370	15,4	48,38	186,27	154
155	24 025	3 723 875	12,4499	5,3717	0,00645	5,0434	15,5	48,69	188,69	155
156	24 336	3 796 416	12,4900	5,3832	0,00641	5,0499	15,6	49,01	191,13	156
157	24 649	3 869 893	12,5300	5,3947	0,00637	5,0562	15,7	49,32	193,59	157
158	24 964	3 944 312	12,5698	5,4061	0,00633	5,0626	15,8	49,64	196,07	158
159	25 281	4 019 679	12,6095	5,4175	0,00629	5,0689	15,9	49,95	198,56	159
160	25 600	4 096 000	12,6491	5,4288	0,00625	5,0752	16,0	50,27	201,06	**160**
161	25 921	4 173 281	12,6886	5,4401	0,00621	5,0814	16,1	50,58	203,58	161
162	26 244	4 251 528	12,7279	5,4514	0,00617	5,0876	16,2	50,89	206,12	162
163	26 569	4 330 747	12,7671	5,4626	0,00613	5,0938	16,3	51,21	208,67	163
164	26 896	4 410 944	12,8062	5,4737	0,00610	5,0999	16,4	51,52	211,24	164
165	27 225	4 492 125	12,8452	5,4848	0,00606	5,1059	16,5	51,84	213,82	165
166	27 556	4 574 296	12,8841	5,4959	0,00602	5,1120	16,6	52,15	216,42	166
167	27 889	4 657 463	12,9228	5,5069	0,00599	5,1180	16,7	52,46	219,04	167
168	28 224	4 741 632	12,9615	5,5178	0,00595	5,1240	16,8	52,78	221,67	168
169	28 561	4 826 809	13,0000	5,5288	0,00592	5,1299	16,9	53,09	224,32	169
170	28 900	4 913 000	13,0384	5,5397	0,00588	5,1358	17,0	53,41	226,98	**170**
171	29 241	5 000 211	13,0767	5,5505	0,00585	5,1417	17,1	53,72	229,66	171
172	29 584	5 088 448	13,1149	5,5613	0,00581	5,1475	17,2	54,04	232,35	172
173	29 929	5 177 717	13,1529	5,5721	0,00578	5,1533	17,3	54,35	235,06	173

174	30 276	5 268 024	13,1909	5,5828	0,00575	5,1591	17,4	54,66	237,79	174
175	30 625	5 359 375	13,2288	5,5934	0,00571	5,1648	17,5	54,98	240,53	175
176	30 976	5 451 776	13,2665	5,6041	0,00568	5,1705	17,6	55,29	243,28	176
177	31 329	5 545 233	13,3041	5,6147	0,00565	5,1761	17,7	55,61	246,06	177
178	31 684	5 639 752	13,3417	5,6252	0,00562	5,1818	17,8	55,92	248,85	178
179	32 041	5 735 339	13,3791	5,6357	0,00559	5,1874	17,9	56,23	251,65	179
180	32 400	5 832 000	13,4164	5,6462	0,00556	5,1930	18,0	56,55	254,47	180
181	32 761	5 929 741	13,4536	5,6567	0,00552	5,1985	18,1	56,86	257,30	181
182	33 124	6 028 568	13,4907	5,6671	0,00549	5,2040	18,2	57,18	260,16	182
183	33 489	6 128 487	13,5277	5,6774	0,00546	5,2095	18,3	57,49	263,02	183
184	33 856	6 229 504	13,5647	5,6877	0,00543	5,2149	18,4	57,81	265,90	184
185	34 225	6 331 625	13,6015	5,6980	0,00541	5,2204	18,5	58,12	268,80	185
186	34 596	6 434 856	13,6382	5,7083	0,00538	5,2257	18,6	58,43	271,72	186
187	34 969	6 539 203	13,6748	5,7185	0,00535	5,2311	18,7	58,75	274,65	187
188	35 344	6 644 672	13,7113	5,7287	0,00532	5,2364	18,8	59,06	277,59	188
189	35 721	6 751 269	13,7477	5,7388	0,00529	5,2417	18,9	59,38	280,55	189
190	36 100	6 859 000	13,7840	5,7489	0,00526	5,2470	19,0	59,69	283,53	190
191	36 481	6 967 871	13,8203	5,7590	0,00524	5,2523	19,1	60,00	286,52	191
192	36 864	7 077 888	13,8564	5,7690	0,00521	5,2575	19,2	60,32	289,53	192
193	37 249	7 189 057	13,8924	5,7790	0,00518	5,2627	19,3	60,63	292,55	193
194	37 636	7 301 384	13,9284	5,7890	0,00515	5,2679	19,4	60,95	295,59	194
195	38 025	7 414 875	13,9642	5,7989	0,00513	5,2730	19,5	61,26	298,65	195
196	38 416	7 529 536	14,0000	5,8088	0,00510	5,2781	19,6	61,58	301,72	196
197	38 809	7 645 373	14,0357	5,8186	0,00508	5,2832	19,7	61,89	304,81	197
198	39 204	7 762 392	14,0712	5,8285	0,00505	5,2883	19,8	62,20	307,91	198
199	39 601	7 880 599	14,1067	5,8383	0,00503	5,2933	19,9	62,52	311,03	199
200	40 000	8 000 000	14,1421	5,8480	0,00500	5,2983	20,0	62,83	314,16	200

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1n	πd	$\frac{1}{4}\pi d^2$	n
201	40 401	8 120 601	14,1774	5,8578	0,00498	5,3033	20,1	63,15	317,31	201
202	40 804	8 242 408	14,2127	5,8675	0,00495	5,3083	20,2	63,46	320,47	202
203	41 209	8 365 427	14,2478	5,8771	0,00493	5,3132	20,3	63,77	323,65	203
204	41 616	8 489 664	14,2829	5,8868	0,00490	5,3181	20,4	64,09	326,85	204
205	42 025	8 615 125	14,3178	5,8964	0,00488	5,3230	20,5	64,40	330,06	205
206	42 436	8 741 816	14,3527	5,9059	0,00485	5,3279	20,6	64,72	333,29	206
207	42 849	8 869 743	14,3875	5,9155	0,00483	5,3327	20,7	65,03	336,54	207
208	43 264	8 998 912	14,4222	5,9250	0,00481	5,3375	20,8	65,35	339,79	208
209	43 681	9 129 329	14,4568	5,9345	0,00478	5,3423	20,9	65,66	343,07	209
210	44 100	9 261 000	14,4914	5,9439	0,00476	5,3471	21,0	65,97	346,36	210
211	44 521	9 393 931	14,5258	5,9533	0,00474	5,3519	21,1	66,29	349,67	211
212	44 944	9 528 128	14,5602	5,9627	0,00472	5,3566	21,2	66,60	352,99	212
213	45 369	9 663 597	14,5945	5,9721	0,00469	5,3613	21,3	66,92	356,33	213
214	45 796	9 800 344	14,6287	5,9814	0,00467	5,3660	21,4	67,23	359,68	214
215	46 225	9 938 375	14,6629	5,9907	0,00465	5,3706	21,5	67,54	363,05	215
216	46 656	10 077 696	14,6969	6,0000	0,00463	5,3753	21,6	67,86	366,44	216
217	47 089	10 218 313	14,7309	6,0092	0,00461	5,3799	21,7	68,17	369,84	217
218	47 524	10 360 232	14,7648	6,0185	0,00459	5,3845	21,8	68,49	373,25	218
219	47 961	10 503 459	14,7986	6,0277	0,00457	5,3891	21,9	68,80	376,68	219
220	48 400	10 648 000	14,8324	6,0368	0,00455	5,3936	22,0	69,12	380,13	220
221	48 841	10 793 861	14,8661	6,0459	0,00452	5,3982	22,1	69,43	383,60	221
222	49 284	10 941 048	14,8997	6,0550	0,00450	5,4027	22,2	69,74	387,08	222
223	49 729	11 089 567	14,9332	6,0641	0,00448	5,4072	22,3	70,06	390,57	223

224	50 176	11 239 424	14,9666	6,0732	0,00446	5,4116	22,4	70,37	394,08	224
225	50 625	11 390 625	15,0000	6,0822	0,00444	5,4161	22,5	70,69	397,61	225
226	51 076	11 543 176	15,0333	6,0912	0,00442	5,4205	22,6	71,00	401,15	226
227	51 529	11 697 083	15,0665	6,1002	0,00441	5,4250	22,7	71,31	404,71	227
228	51 984	11 852 352	15,0997	6,1091	0,00439	5,4293	22,8	71,63	408,28	228
229	52 441	12 008 989	15,1327	6,1180	0,00437	5,4337	22,9	71,94	411,87	229
230	52 900	12 167 000	15,1658	6,1269	0,00435	5,4381	23,0	72,26	415,48	230
231	53 361	12 326 391	15,1987	6,1358	0,00433	5,4424	23,1	72,57	419,10	231
232	53 824	12 487 168	15,2315	6,1446	0,00431	5,4467	23,2	72,88	422,73	232
233	54 289	12 649 337	15,2643	6,1534	0,00429	5,4510	23,3	73,20	426,38	233
234	54 756	12 812 904	15,2971	6,1622	0,00427	5,4553	23,4	73,51	430,05	234
235	55 225	12 977 875	15,3297	6,1710	0,00426	5,4596	23,5	73,83	433,74	235
236	55 696	13 144 256	15,3623	6,1797	0,00424	5,4638	23,6	74,14	437,44	236
237	56 169	13 312 053	15,3948	6,1885	0,00422	5,4681	23,7	74,46	441,15	237
238	56 644	13 481 272	15,4272	6,1972	0,00420	5,4723	23,8	74,77	444,88	238
239	57 121	13 651 919	15,4596	6,2058	0,00418	5,4765	23,9	75,08	448,63	239
240	57 600	13 824 000	15,4919	6,2145	0,00417	5,4806	24,0	75,40	452,39	240
241	58 081	13 997 521	15,5242	6,2231	0,00415	5,4848	24,1	75,71	456,17	241
242	58 564	14 172 488	15,5563	6,2317	0,00413	5,4889	24,2	76,03	459,96	242
243	59 049	14 348 907	15,5885	6,2403	0,00412	5,4931	24,3	76,34	463,77	243
244	59 536	14 526 784	15,6205	6,2488	0,00410	5,4972	24,4	76,65	467,59	244
245	60 025	14 706 125	15,6525	6,2573	0,00408	5,5013	24,5	76,97	471,44	245
246	60 516	14 886 936	15,6844	6,2658	0,00407	5,5053	24,6	77,28	475,29	246
247	61 009	15 069 223	15,7162	6,2743	0,00405	5,5094	24,7	77,60	479,16	247
248	61 504	15 252 992	15,7480	6,2828	0,00403	5,5134	24,8	77,91	483,05	248
249	62 001	15 438 249	15,7797	6,2912	0,00402	5,5175	24,9	78,23	486,95	249
250	62 500	15 625 000	15,8114	6,2996	0,00400	5,5215	25,0	78,54	490,87	250

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
251	63 001	15 813 251	15,8430	6,3080	0,00398	5,5255	25,1	78,85	494,81	251
252	63 504	16 003 008	15,8745	6,3164	0,00397	5,5294	25,2	79,17	498,76	252
253	64 009	16 194 277	15,9060	6,3247	0,00395	5,5334	25,3	79,48	502,78	253
254	64 516	16 387 064	15,9374	6,3330	0,00394	5,5373	25,4	79,80	506,71	254
255	65 025	16 581 375	15,9687	6,3413	0,00392	5,5413	25,5	80,11	510,71	255
256	65 536	16 777 216	16,0000	6,3496	0,00391	5,5452	25,6	80,42	514,72	256
257	66 049	16 974 593	16,0312	6,3579	0,00389	5,5491	25,7	80,74	518,75	257
258	66 564	17 173 512	16,0624	6,3661	0,00388	5,5530	25,8	81,05	522,79	258
259	67 081	17 373 979	16,0935	6,3743	0,00386	5,5568	25,9	81,37	526,85	259
260	67 600	17 576 000	16,1245	6,3825	0,00385	5,5607	26,0	81,68	530,93	260
261	68 121	17 779 581	16,1555	6,3907	0,00383	5,5645	26,1	82,00	535,02	261
262	68 644	17 984 728	16,1864	6,3988	0,00382	5,5683	26,2	82,31	539,13	262
263	69 169	18 191 447	16,2173	6,4070	0,00380	5,5722	26,3	82,62	543,25	263
264	69 696	18 399 744	16,2481	6,4151	0,00379	5,5759	26,4	82,94	547,39	264
265	70 225	18 609 625	16,2788	6,4232	0,00377	5,5797	26,5	83,25	551,55	265
266	70 756	18 821 096	16,3095	6,4312	0,00376	5,5835	26,6	83,57	555,72	266
267	71 289	19 034 163	16,3401	6,4393	0,00375	5,5872	26,7	83,88	559,90	267
268	71 824	19 248 832	16,3707	6,4473	0,00373	5,5910	26,8	84,19	564,10	268
269	72 361	19 465 109	16,4012	6,4553	0,00372	5,5947	26,9	84,51	568,32	269
270	72 900	19 683 000	16,4317	6,4633	0,00370	5,5984	27,0	84,82	572,56	270
271	73 441	19 902 511	16,4621	6,4713	0,00369	5,6021	27,1	85,14	576,80	271
272	73 984	20 123 648	16,4924	6,4792	0,00368	5,6058	27,2	85,45	581,07	272
273	74 529	20 346 417	16,5227	6,4872	0,00366	5,6095	27,3	85,77	585,35	273

274	75 076	20 570 824	16,5529	6,4951	0,00365	5,6181	27,4	86,08	589,65	274
275	75 625	20 796 875	16,5881	6,5030	0,00364	5,6168	27,5	86,39	593,96	275
276	76 176	21 024 576	16,6132	6,5108	0,00362	5,6204	27,6	86,71	598,28	276
277	76 729	21 253 933	16,6483	6,5187	0,00361	5,6240	27,7	87,02	602,63	277
278	77 284	21 484 952	16,6733	6,5265	0,00360	5,6276	27,8	87,34	606,99	278
279	77 841	21 717 639	16,7083	6,5343	0,00358	5,6312	27,9	87,65	611,36	279
280	78 400	21 952 000	16,7332	6,5421	0,00357	5,6348	28,0	87,96	615,75	**280**
281	78 961	22 188 041	16,7631	6,5499	0,00356	5,6384	28,1	88,28	620,16	281
282	79 524	22 425 768	16,7929	6,5577	0,00355	5,6419	28,2	88,59	624,58	282
283	80 089	22 665 187	16,8226	6,5654	0,00353	5,6454	28,3	88,91	629,02	283
284	80 656	22 906 304	16,8523	6,5731	0,00352	5,6490	28,4	89,22	633,47	284
285	81 225	23 149 125	16,8819	6,5808	0,00351	5,6525	28,5	89,54	637,94	285
286	81 796	23 393 656	16,9115	6,5885	0,00350	5,6560	28,6	89,85	642,42	286
287	82 369	23 639 903	16,9411	6,5962	0,00348	5,6595	28,7	90,16	646,92	287
288	82 944	23 887 872	16,9706	6,6039	0,00347	5,6630	28,8	90,48	651,44	288
289	83 521	24 137 569	17,0000	6,6115	0,00346	5,6664	28,9	90,79	655,97	289
290	84 100	24 389 000	17,0294	6,6191	0,00345	5,6699	29,0	91,11	660,52	**290**
291	84 681	24 642 171	17,0587	6,6267	0,00344	5,6733	29,1	91,42	665,08	291
292	85 264	24 897 088	17,0880	6,6343	0,00342	5,6768	29,2	91,73	669,66	292
293	85 849	25 153 757	17,1172	6,6419	0,00341	5,6802	29,3	92,05	674,26	293
294	86 436	25 412 184	17,1464	6,6494	0,00340	5,6836	29,4	92,36	678,87	294
295	87 025	25 672 375	17,1756	6,6569	0,00339	5,6870	29,5	92,68	683,49	295
296	87 616	25 934 336	17,2047	6,6644	0,00338	5,6904	29,6	92,99	688,13	296
297	88 209	26 198 073	17,2337	6,6719	0,00337	5,6937	29,7	93,31	692,79	297
298	88 804	26 463 592	17,2627	6,6794	0,00336	5,6971	29,8	93,62	697,46	298
299	89 401	26 730 899	17,2916	6,6869	0,00334	5,7004	29,9	93,93	702,15	299
300	90 000	27 000 000	17,3205	6,6943	0,00333	5,7038	30,0	94,25	706,86	**300**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1n	πd	$\frac{1}{4}\pi d^2$	n
301	90 601	27 270 901	17,3494	6,7018	0,00332	5,7071	30,1	94,56	711,58	301
302	91 204	27 543 608	17,3781	6,7092	0,00331	5,7104	30,2	94,88	716,31	302
303	91 809	27 818 127	17,4069	6,7166	0,00330	5,7137	30,3	95,19	721,07	303
304	92 416	28 094 464	17,4356	6,7240	0,00329	5,7170	30,4	95,50	725,83	304
305	93 025	28 372 625	17,4642	6,7313	0,00328	5,7203	30,5	95,82	730,62	305
306	93 636	28 652 616	17,4929	6,7387	0,00327	5,7236	30,6	96,13	735,42	306
307	94 249	28 934 443	17,5214	6,7460	0,00326	5,7268	30,7	96,45	740,23	307
308	94 864	29 218 112	17,5499	6,7533	0,00325	5,7301	30,8	96,76	745,06	308
309	95 481	29 503 629	17,5784	6,7606	0,00324	5,7333	30,9	97,08	749,91	309
310	96 100	29 791 000	17,6068	6,7679	0,00323	5,7366	31,0	97,39	754,77	**310**
311	96 721	30 080 231	17,6352	6,7752	0,00322	5,7398	31,1	97,70	759,64	311
312	97 344	30 371 328	17,6635	6,7824	0,00321	5,7430	31,2	98,02	764,54	312
313	97 969	30 664 297	17,6918	6,7897	0,00319	5,7462	31,3	98,33	769,44	313
314	98 596	30 959 144	17,7200	6,7969	0,00318	5,7494	31,4	98,65	774,37	314
315	99 225	31 255 875	17,7482	6,8041	0,00317	5,7526	31,5	98,96	779,31	315
316	99 856	31 554 496	17,7764	6,8113	0,00316	5,7557	31,6	99,27	784,27	316
317	100 489	31 855 013	17,8045	6,8185	0,00315	5,7589	31,7	99,59	789,24	317
318	101 124	32 157 432	17,8326	6,8256	0,00314	5,7621	31,8	99,90	794,23	318
319	101 761	32 461 759	17,8606	6,8328	0,00313	5,7652	31,9	100,2	799,23	319
320	102 400	32 768 000	17,8885	6,8399	0,00313	5,7683	32,0	100,5	804,25	**320**
321	103 041	33 076 161	17,9165	6,8470	0,00312	5,7714	32,1	100,8	809,28	321
322	103 684	33 386 248	17,9444	6,8541	0,00311	5,7746	32,2	101,2	814,33	322
323	104 329	33 698 267	17,9722	6,8612	0,00310	5,7777	32,3	101,5	819,40	323

324	104 976	34 012 224	18,0000	6,8683	0,00309	5,7807	32,4	101,8	824,48	324
325	105 625	34 328 125	18,0278	6,8753	0,00308	5,7838	32,5	102,1	829,58	325
326	106 276	34 645 976	18,0555	6,8824	0,00307	5,7869	32,6	102,4	834,69	326
327	106 929	34 965 783	18,0831	6,8894	0,00306	5,7900	32,7	102,7	839,82	327
328	107 584	35 287 552	18,1108	6,8964	0,00305	5,7930	32,8	103,0	844,96	328
329	108 241	35 611 289	18,1384	6,9034	0,00304	5,7961	32,9	103,4	850,12	329
330	108 900	35 937 000	18,1659	6,9104	0,00303	5,7991	33,0	103,7	855,30	**330**
331	109 561	36 264 691	18,1934	6,9174	0,00302	5,8021	33,1	104,0	860,49	331
332	110 224	36 594 368	18,2209	6,9244	0,00301	5,8051	33,2	104,3	865,70	332
333	110 889	36 926 037	18,2483	6,9313	0,00300	5,8081	33,3	104,6	870,92	333
334	111 556	37 259 704	18,2757	6,9382	0,00299	5,8111	33,4	104,9	876,16	334
335	112 225	37 595 375	18,3030	6,9451	0,00299	5,8141	33,5	105,2	881,41	335
336	112 896	37 933 056	18,3303	6,9521	0,00298	5,8171	33,6	105,6	886,68	336
337	113 569	38 272 753	18,3576	6,9589	0,00297	5,8201	33,7	105,9	891,97	337
338	114 244	38 614 472	18,3848	6,9658	0,00296	5,8230	33,8	106,2	897,27	338
339	114 921	38 958 219	18,4120	6,9727	0,00295	5,8260	33,9	106,5	902,59	339
340	115 600	39 304 000	18,4391	6,9795	0,00294	5,8289	34,0	106,8	907,92	**340**
341	116 281	39 651 821	18,4662	6,9864	0,00293	5,8319	34,1	107,1	913,27	341
342	116 964	40 001 688	18,4932	6,9932	0,00292	5,8348	34,2	107,4	918,63	342
343	117 649	40 353 607	18,5203	7,0000	0,00292	5,8377	34,3	107,8	924,01	343
344	118 336	40 707 584	18,5472	7,0068	0,00291	5,8406	34,4	108,1	929,41	344
345	119 025	41 063 625	18,5742	7,0136	0,00290	5,8435	34,5	108,4	934,82	345
346	119 716	41 421 736	18,6011	7,0203	0,00289	5,8464	34,6	108,7	940,25	346
347	120 409	41 781 923	18,6279	7,0271	0,00288	5,8493	34,7	109,0	945,69	347
348	121 104	42 144 192	18,6548	7,0338	0,00287	5,8522	34,8	109,3	951,15	348
349	121 801	42 508 549	18,6815	7,0406	0,00287	5,8551	34,9	109,6	956,62	349
350	122 500	42 875 000	18,7083	7,0473	0,00286	5,8579	35,0	110,0	962,11	**350**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
351	123 201	43 243 551	18,7350	7,0540	0,00285	5,8608	35,1	110,3	967,62	351
352	123 904	43 614 208	18,7617	7,0607	0,00284	5,8636	35,2	110,6	973,14	352
353	124 609	43 986 977	18,7883	7,0674	0,00283	5,8665	35,3	110,9	978,68	353
354	125 316	44 361 864	18,8149	7,0740	0,00282	5,8693	35,4	111,2	984,23	354
355	126 025	44 738 875	18,8414	7,0807	0,00282	5,8721	35,5	111,5	989,80	355
356	126 736	45 118 016	18,8680	7,0873	0,00281	5,8749	35,6	111,8	995,38	356
357	127 449	45 499 293	18,8944	7,0940	0,00280	5,8777	35,7	112,2	1001,0	357
358	128 164	45 882 712	18,9209	7,1006	0,00279	5,8805	35,8	112,5	1006,6	358
359	128 881	46 268 279	18,9473	7,1072	0,00279	5,8833	35,9	112,8	1012,2	359
360	129 600	46 656 000	18,9737	7,1138	0,00278	5,8861	36,0	113,1	1017,9	360
361	130 321	47 045 881	19,0000	7,1204	0,00277	5,8889	36,1	113,4	1023,5	361
362	131 044	47 437 928	19 0263	7,1269	0,00276	5,8916	36,2	113,7	1029,2	362
363	131 769	47 832 147	19,0526	7,1335	0,00275	5,8944	36,3	114,0	1034,9	363
364	132 496	48 228 544	19,0788	7,1400	0,00275	5,8972	36,4	114,4	1040,6	364
365	133 225	48 627 125	19,1050	7,1466	0,00274	5,8999	36,5	114,7	1046,3	365
366	133 956	49 027 896	19,1311	7,1531	0,00273	5,9026	36,6	115,0	1052,1	366
367	134 689	49 430 863	19,1572	7,1596	0,00272	5,9054	36,7	115,3	1057,8	367
368	135 424	49 836 032	19,1833	7,1661	0,00272	5,9081	36,8	115,6	1063,6	368
369	136 161	50 243 409	19,2094	7,1726	0,00271	5,9108	36,9	115,9	1069,4	369
370	136 900	50 653 000	19,2354	7,1791	0,00270	5,9135	37,0	116,2	1075,2	370
371	137 641	51 064 811	19,2614	7,1855	0,00270	5,9162	37,1	116,6	1081,0	371
372	138 384	51 478 848	19,2873	7,1920	0,00269	5,9189	37,2	116,9	1086,9	372
373	139 129	51 895 117	19,3132	7,1984	0,00268	5,9216	37,3	117,2	1092,7	373

374	139 876	52 313 624	19,3391	7,2048	0,00267	5,9243	37,4	117,5	1098,6	374
375	140 625	52 734 375	19,3649	7,2112	0,00267	5,9269	37,5	117,8	1104,5	375
376	141 376	53 157 376	19,3907	7,2177	0,00266	5,9296	37,6	118,1	1110,4	376
377	142 129	53 582 633	19,4165	7,2240	0,00265	5,9322	37,7	118,4	1116,3	377
378	142 884	54 010 152	19,4422	7,2304	0,00265	5,9349	37,8	118,8	1122,2	378
379	143 641	54 439 939	19,4679	7,2368	0,00264	5,9375	37,9	119,1	1128,2	379
380	144 400	54 872 000	19,4936	7,2432	0,00263	5,9402	38,0	119,4	1134,1	**380**
381	145 161	55 306 341	19,5192	7,2495	0,00262	5,9428	38,1	119,7	1140,1	381
382	145 924	55 742 968	19,5448	7,2558	0,00262	5,9454	38,2	120,0	1146,1	382
383	146 689	56 181 887	19,5704	7,2622	0,00261	5,9480	38,3	120,3	1152,1	383
384	147 456	56 623 104	19,5959	7,2685	0,00260	5,9506	38,4	120,6	1158,1	384
385	148 225	57 066 625	19,6214	7,2748	0,00260	5,9532	38,5	121,0	1164,2	385
386	148 996	57 512 456	19,6469	7,2811	0,00259	5,9558	38,6	121,3	1170,2	386
387	149 769	57 960 603	19,6723	7,2874	0,00258	5,9584	38,7	121,6	1176,3	387
388	150 544	58 411 072	19,6977	7,2936	0,00258	5,9610	38,8	121,9	1182,4	388
389	151 321	58 863 869	19,7231	7,2999	0,00257	5,9636	38,9	122,2	1188,5	389
390	152 100	59 319 000	19,7484	7,3061	0,00256	5,9661	39,0	122,5	1194,6	**390**
391	152 881	59 776 471	19,7737	7,3124	0,00256	5,9687	39,1	122,8	1200,7	391
392	153 664	60 236 288	19,7990	7,3186	0,00255	5,9713	39,2	123,2	1206,9	392
393	154 449	60 698 457	19,8242	7,3248	0,00254	5,9738	39,3	123,5	1213,0	393
394	155 236	61 162 984	19,8494	7,3310	0,00254	5,9764	39,4	123,8	1219,2	394
395	156 025	61 629 875	19,8746	7,3372	0,00253	5,9789	39,5	124,1	1225,4	395
396	156 816	62 099 136	19,8997	7,3434	0,00253	5,9814	39,6	124,4	1231,6	396
397	157 609	62 570 773	19,9249	7,3496	0,00252	5,9839	39,7	124,7	1237,9	397
398	158 404	63 044 792	19 9499	7,3558	0,00251	5,9865	39,8	125,0	1244,1	398
399	159 201	63 521 199	19,9750	7,3619	0,00251	5,9890	39,9	125,3	1250,4	399
400	160 000	64 000 000	20,0000	7,3681	0,00250	5,9914	40,0	125,7	1256,6	**400**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
401	160 801	64 481 201	20,0250	7,3742	0,00249	5,9940	40,1	126,0	1262,9	401
402	161 604	64 964 808	20,0499	7,3803	0,00249	5,9965	40,2	126,3	1269,2	402
403	162 409	65 450 827	20,0749	7,3864	0,00248	5,9989	40,3	126,6	1275,6	403
404	163 216	65 939 264	20,0998	7,3925	0,00248	6,0014	40,4	126,9	1281,9	404
405	164 025	66 430 125	20,1246	7,3986	0,00247	6,0039	40,5	127,2	1288,2	405
406	164 836	66 923 416	20,1494	7,4047	0,00246	6,0064	40,6	127,5	1294,6	406
407	165 649	67 419 143	20,1742	7,4108	0,00246	6,0088	40,7	127,9	1301,0	407
408	166 464	67 917 312	20,1990	7,4169	0,00245	6,0113	40,8	128,2	1307,4	408
409	167 281	68 417 929	20,2237	7,4229	0,00244	6,0137	40,9	128,5	1313,8	409
410	168 100	68 921 000	20.2485	7,4290	0,00244	6,0162	41,0	128,8	1320,3	410
411	168 921	69 426 531	20,2731	7,4350	0,00243	6,0186	41,1	129,1	1326,7	411
412	169 744	69 934 528	20,2978	7,4410	0,00243	6,0210	41,2	129,4	1333,2	412
413	170 569	70 444 997	20,3224	7,4470	0,00242	6,0234	41,3	129,7	1339,6	413
414	171 396	70 957 944	20,3470	7,4530	0,00242	6,0259	41,4	130,1	1346,1	414
415	172 225	71 473 375	20,3715	7,4590	0,00241	6,0283	41,5	130,4	1352,7	415
416	173 056	71 991 296	20,3961	7,4650	0,00240	6,0307	41,6	130,7	1359,2	416
417	173 889	72 511 713	20,4206	7,4710	0,00240	6,0331	41,7	131,0	1365,7	417
418	174 724	73 034 632	20,4450	7,4770	0.00239	6,0355	41,8	131,3	1372,3	418
419	175 561	73 560 059	20,4695	7,4829	0,00239	6,0379	41,9	131,6	1378,9	419
420	176 400	74 088 000	20,4939	7,4889	0,00238	6,0403	42,0	131,9	1385,4	420
421	177 241	74 618 461	20,5183	7,4948	0,00238	6,0426	42,1	132,3	1392,0	421
422	178 084	75 151 448	20,5426	7,5007	0,00237	6,0450	42,2	132,6	1398,7	422
423	178 929	75 686 967	20,5670	7,5067	0,00236	6,0474	42,3	132,9	1405,3	423

424	179 776	76 225 024	20,5913	7,5126	0,00236	6,0497	42,4	133,2	1412,0	424
425	180 625	76 765 625	20,6155	7,5185	0,00235	6,0521	42,5	133,5	1418,6	425
426	181 476	77 308 776	20,6398	7,5244	0,00235	6,0544	42,6	133,8	1425,3	426
427	182 329	77 854 483	20,6640	7,5302	0,00234	6,0568	42,7	134,1	1432,0	427
428	183 184	78 402 752	20,6882	7,5361	0,00234	6,0591	42,8	134,5	1438,7	428
429	184 041	78 953 589	20,7123	7,5420	0,00233	6,0615	42,9	134,8	1445,5	429
430	184 900	79 507 000	20,7364	7,5478	0,00233	6,0638	43,0	135,1	1452,2	430
431	185 761	80 062 991	20,7605	7,5537	0,00232	6,0661	43,1	135,4	1459,0	431
432	186 624	80 621 568	20,7846	7,5595	0,00231	6,0684	43,2	135,7	1465,7	432
433	187 489	81 182 737	20,8087	7,5654	0,00231	6,0707	43,3	136,0	1472,5	433
434	188 356	81 746 504	20,8327	7,5712	0,00230	6,0730	43,4	136,3	1479,3	434
435	189 225	82 312 875	20,8567	7,5770	0,00230	6,0753	43,5	136,7	1486,2	435
436	190 096	82 881 856	20,8806	7,5828	0,00229	6,0776	43,6	137,0	1493,0	436
437	190 969	83 453 453	20,9045	7,5886	0,00229	6,0799	43,7	137,3	1499,9	437
438	191 844	84 027 672	20,9284	7,5944	0,00228	6,0822	43,8	137,6	1506,7	438
439	192 721	84 604 519	20,9523	7,6001	0,00228	6,0845	43,9	137,9	1513,6	439
440	193 600	85 184 000	20,9762	7,6059	0,00227	6,0868	44,0	138,2	1520,5	440
441	194 481	85 766 121	21,0000	7,6117	0,00227	6,0890	44,1	138,5	1527,5	441
442	195 364	86 350 888	21,0238	7,6174	0,00226	6,0913	44,2	138,9	1534,4	442
443	196 249	86 938 307	21,0476	7,6232	0,00226	6,0936	44,3	139,2	1541,3	443
444	197 136	87 528 384	21,0713	7,6289	0,00225	6,0958	44,4	139,5	1548,3	444
445	198 025	88 121 125	21,0950	7,6346	0,00225	6,0981	44,5	139,8	1555,3	445
446	198 916	88 716 536	21,1187	7,6403	0,00224	6,1003	44,6	140,1	1562,3	446
447	199 809	89 314 623	21,1424	7,6460	0,00224	6,1026	44,7	140,4	1569,3	447
448	200 704	89 915 392	21,1660	7,6517	0,00223	6,1048	44,8	140,7	1576,3	448
449	201 601	90 518 849	21,1896	7,6574	0,00223	6,1070	44,9	141,1	1583,4	449
450	202 500	91 125 000	21,2132	7,6631	0,00222	6,1092	45,0	141,4	1590,4	450

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
451	203 401	91 733 851	21,2368	7,6688	0,00222	6,1115	45,1	141,7	1597,5	451
452	204 304	92 345 408	21,2603	7,6744	0,00221	6,1137	45,2	142.0	1604,6	452
453	205 209	92 959 677	21,2838	7,6801	0,00221	6,1159	45,3	142,3	1611,7	453
454	206 116	93 576 664	21,3073	7,6857	0,00220	6,1181	45,4	142,6	1618,8	454
455	207 025	94 196 375	21,3307	7,6914	0,00220	6,1203	45,5	142,9	1626,0	455
456	207 936	94 818 816	21,3542	7,6970	0,00219	6,1225	45,6	143,3	1633,1	456
457	208 849	95 443 993	21,3776	7,7026	0,00219	6,1247	45,7	143,6	1640,3	457
458	209 764	96 071 912	21,4009	7,7082	0,00218	6,1269	45,8	143,9	1647,5	458
459	210 681	96 702 579	21,4243	7,7138	0,00218	6,1291	45,9	144,2	1654,7	459
460	211 600	97 336 000	21,4476	7,7194	0,00217	6,1312	46,0	144,5	1661,9	**460**
461	212 521	97 972 181	21,4709	7,7250	0,00217	6,1334	46,1	144,8	1669,1	461
462	213 444	98 611 128	21,4942	7,7306	0,00216	6,1356	46,2	145,1	1676,4	462
463	214 369	99 252 847	21,5174	7,7362	0,00216	6,1377	46,3	145,5	1683,7	463
464	215 296	99 897 344	21,5407	7,7418	0,00216	6,1399	46,4	145,8	1690,9	464
465	216 225	100 544 625	21,5639	7,7473	0,00215	6,1420	46,5	146,1	1698,2	465
466	217 156	101 194 696	21,5870	7,7529	0,00215	6,1442	46,6	146,4	1705,5	466
467	218 089	101 847 563	21,6102	7,7584	0,00214	6,1463	46,7	146,7	1712,9	467
468	219 024	102 503 232	21,6333	7,7639	0,00214	6,1485	46,8	147,0	1720,2	468
469	219 961	103 161 709	21,6564	7,7695	0,00213	6,1506	46,9	147,3	1727,6	469
470	220 900	103 823 000	21,6795	7,7750	0,00213	6,1527	47,0	147,7	1734,9	**470**
471	221 841	104 487 111	21,7025	7,7805	0,00212	6,1549	47,1	148,0	1742,3	471
472	222 784	105 154 048	21,7256	7,7860	0,00212	6,1570	47,2	148,3	1749,7	472
473	223 729	105 823 817	21,7486	7,7915	0,00211	6,1591	47,3	148,6	1757,2	473

474	224 676	106 496 424	21,7715	7,7970	0,00211	6,1612	47,4	148,9	1764,6	474
475	225 625	107 171 875	21,7945	7,8025	0,00211	6,1638	47,5	149,2	1772,1	475
476	226 576	107 850 176	21,8174	7,8079	0,00210	6,1654	47,6	149,5	1779,5	476
477	227 529	108 531 333	21,8403	7,8134	0,00210	6,1675	47,7	149,9	1787,0	477
478	228 484	109 215 352	21,8632	7,8188	0,00209	6,1696	47,8	150,2	1794,5	478
479	229 441	109 902 239	21,8861	7,8243	0,00209	6,1717	47,9	150,5	1802,0	479
480	230 400	110 592 000	21,9089	7,8297	0,00208	6,1738	48,0	150,8	1809,6	480
481	231 361	111 284 641	21,9317	7,8352	0,00208	6,1759	48,1	151,1	1817,1	481
482	232 324	111 980 168	21,9545	7,8406	0,00207	6,1779	48,2	151,4	1824,7	482
483	233 289	112 678 587	21,9773	7,8460	0,00207	6,1800	48,3	151,7	1832,2	483
484	234 256	113 379 904	22,0000	7,8514	0,00207	6,1821	48,4	152,1	1839,8	484
485	235 225	114 084 125	22,0227	7,8568	0,00206	6,1841	48,5	152,4	1847,5	485
486	236 196	114 791 256	22,0454	7,8622	0,00206	6,1862	48,6	152,7	1855,1	486
487	237 169	115 501 303	22,0681	7,8676	0,00205	6,1883	48,7	153,0	1862,7	487
488	238 144	116 214 272	22,0907	7,8730	0,00205	6,1903	48,8	153,3	1870,4	488
489	239 121	116 930 169	22,1133	7,8784	0,00204	6,1924	48,9	153,6	1878,1	489
490	240 100	117 649 000	22,1359	7,8837	0,00204	6,1944	49,0	153,9	1885,7	490
491	241 081	118 370 771	22,1585	7,8891	0,00204	6,1964	49,1	154,3	1893,4	491
492	242 064	119 095 488	22,1811	7,8944	0,00203	6,1985	49,2	154,6	1901,2	492
493	243 049	119 823 157	22,2036	7,8998	0,00203	6,2005	49,3	154,9	1908,9	493
494	244 036	120 553 784	22,2261	7,9051	0,00202	6,2025	49,4	155,2	1916,7	494
495	245 025	121 287 375	22,2486	7,9105	0,00202	6,2046	49,5	155,5	1924,4	495
496	246 016	122 023 936	22,2711	7,9158	0,00202	6,2066	49,6	155,8	1932,2	496
497	247 009	122 763 473	22,2935	7,9211	0,00201	6,2086	49,7	156,1	1940,0	497
498	248 004	123 505 992	22,3159	7,9264	0,00201	6,2106	49,8	156,5	1947,8	498
499	249 001	124 251 499	22,3383	7,9317	0,00200	6,2126	49,9	156,8	1955,6	499
500	250 000	125 000 000	22,3607	7,9370	0,00200	6,2146	50,0	157,1	1963,5	500

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
501	251 001	125 751 501	22,3830	7,9423	0,00200	6,2166	50,1	157,4	1971,4	501
502	252 004	126 506 008	22,4054	7,9476	0,00199	6,2186	50,2	157,7	1979,2	502
503	253 009	127 263 527	22,4277	7,9528	0,00199	6,2206	50,3	158,0	1987,1	503
504	254 016	128 024 064	22,4499	7,9581	0,00198	6,2226	50,4	158,3	1995,0	504
505	255 025	128 787 625	22,4722	7,9634	0,00198	6,2246	50,5	158,7	2003,0	505
506	256 036	129 554 216	22,4944	7,9686	0,00198	6,2265	50,6	159,0	2010,9	506
507	257 049	130 323 843	22,5167	7,9739	0,00197	6,2285	50,7	159,3	2018,9	507
508	258 064	131 096 512	22,5389	7,9791	0,00197	6,2305	50,8	159,6	2026,8	508
509	259 081	131 872 229	22,5610	7,9843	0,00196	6,2324	50,9	159,9	2034,8	509
510	260 100	132 651 000	22,5832	7,9896	0,00196	6,2344	51,0	160,2	2042,8	510
511	261 121	133 432 831	22,6053	7,9948	0,00196	6,2364	51,1	160,5	2050,8	511
512	262 144	134 217 728	22,6274	8,0000	0,00195	6,2383	51,2	160,8	2058,9	512
513	263 169	135 005 697	22,6495	8,0052	0,00195	6,2403	51,3	161,2	2066,9	513
514	264 196	135 796 744	22,6716	8,0104	0,00195	6,2422	51,4	161,5	2075,0	514
515	265 225	136 590 875	22,6936	8,0156	0,00194	6,2442	51,5	161,8	2083,1	515
516	266 256	137 388 096	22,7156	8,0208	0,00194	6,2461	51,6	162,1	2091,2	516
517	267 289	138 188 413	22,7376	8,0260	0,00193	6,2480	51,7	162,4	2099,3	517
518	268 324	138 991 832	22,7596	8,0311	0,00193	6,2500	51,8	162,7	2107,4	518
519	269 361	139 798 359	22,7816	8,0363	0,00193	6,2519	51,9	163,0	2115,6	519
520	270 400	140 608 000	22,8035	8,0415	0,00192	6,2538	52,0	163,4	2123,7	520
521	271 441	141 420 761	22,8254	8,0466	0,00192	6,2558	52,1	163,7	2131,9	521
522	272 484	142 236 648	22,8473	8,0517	0,00192	6,2577	52,2	164,0	2140,1	522
523	273 529	143 055 667	22,8692	8,0569	0,00191	6,2596	52,3	164,3	2148,3	523

524	274 576	143 877 824	22,8910	8,0620	0,00191	6,2615	52,4	164,6	2156,5	524
525	275 625	144 703 125	22,9129	8,0671	0,00190	6,2634	52,5	164,9	2164,8	525
526	276 676	145 531 576	22,9347	8,0723	0,00190	6,2653	52,6	165,2	2173,0	526
527	277 729	146 363 183	22,9565	8,0774	0,00190	6,2672	52,7	165,6	2181,3	527
528	278 784	147 197 952	22,9783	8,0825	0,00189	6,2691	52,8	165,9	2189,6	528
529	279 841	148 035 889	23,0000	8,0876	0,00189	6,2710	52,9	166,2	2197,9	529
530	280 900	148 877 000	23,0217	8,0927	0,00189	6,2729	53,0	166,5	2206,2	530
531	281 961	149 721 291	23,0434	8,0978	0,00188	6,2748	53,1	166,8	2214,5	531
532	283 024	150 568 768	23,0651	8,1028	0,00188	6,2766	53,2	167,1	2222,9	532
533	284 089	151 419 437	23,0868	8,1079	0,00188	6,2785	53,3	167,4	2231,2	533
534	285 156	152 273 304	23,1084	8,1130	0,00187	6,2804	53,4	167,8	2239,6	534
535	286 225	153 130 375	23,1301	8,1180	0,00187	6,2823	53,5	168,1	2248,0	535
536	287 296	153 990 656	23,1517	8,1231	0,00187	6,2841	53,6	168,4	2256,4	536
537	288 369	154 854 153	23,1733	8,1281	0,00186	6,2860	53,7	168,7	2264,8	537
538	289 444	155 720 872	23,1948	8,1332	0,00186	6,2879	53,8	169,0	2273,3	538
539	290 521	156 590 819	23,2164	8,1382	0,00186	6,2897	53,9	169,3	2281,7	539
540	291 600	157 464 000	23,2379	8,1433	0,00185	6,2916	54,0	169,6	2290,2	540
541	292 681	158 340 421	23,2594	8,1483	0,00185	6,2934	54,1	170,0	2298,7	541
542	293 764	159 220 088	23,2809	8,1533	0,00185	6,2953	54,2	170,3	2307,2	542
543	294 849	160 103 007	23,3024	8,1583	0,00184	6,2971	54,3	170,6	2315,7	543
544	295 936	160 989 184	23,3238	8,1633	0,00184	6,2989	54,4	170,9	2324,3	544
545	297 025	161 878 625	23,3452	8,1683	0,00183	6,3008	54,5	171,2	2332,8	545
546	298 116	162 771 336	23,3666	8,1733	0,00183	6,3026	54,6	171,5	2341,4	546
547	299 209	163 667 323	23,3880	8,1783	0,00183	6,3044	54,7	171,8	2350,0	547
548	300 304	164 566 592	23,4094	8,1833	0,00182	6,3063	54,8	172,2	2358,6	548
549	301 401	165 469 149	23,4307	8,1882	0,00182	6,3081	54,9	172,5	2367,2	549
550	302 500	166 375 000	23,4521	8,1932	0,00182	6,3099	55,0	172,8	2375,8	550

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
551	303 601	167 284 151	23,4734	8,1982	0,00181	6,3117	55,1	173,1	2384,5	551
552	304 704	168 196 608	23,4947	8,2031	0,00181	6,3135	55,2	173,4	2393,1	552
553	305 809	169 112 377	23,5160	8,2081	0,00181	6,3154	55,3	173,7	2401,8	553
554	306 916	170 031 464	23,5372	8,2130	0,00181	6,3172	55,4	174,0	2410,5	554
555	308 025	170 953 875	23,5584	8,2180	0,00180	6,3190	55,5	174,4	2419,2	555
556	309 136	171 879 616	23,5797	8,2229	0,00180	6,3208	55,6	174,7	2427,9	556
557	310 249	172 808 693	23,6008	8,2278	0,00180	6,3226	55,7	175,0	2436,7	557
558	311 364	173 741 112	23,6220	8,2327	0,00179	6,3244	55,8	175,3	2445,4	558
559	312 481	174 676 879	23,6432	8,2377	0,00179	6,3261	55,9	175,6	2454,2	559
560	313 600	175 616 000	23,6643	8,2426	0,00179	6,3279	56,0	175,9	2463,0	560
561	314 721	176 558 481	23,6854	8,2475	0,00178	6,3297	56,1	176,2	2471,8	561
562	315 844	177 504 328	23,7065	8,2524	0,00178	6,3315	56,2	176,6	2480,6	562
563	316 969	178 453 547	23,7276	8,2573	0,00178	6,3333	56,3	176,9	2489,5	563
564	318 096	179 406 144	23,7487	8,2621	0,00177	6,3350	56,4	177,2	2498,3	564
565	319 225	180 362 125	23,7697	8,2670	0,00177	6,3368	56,5	177,5	2507,2	565
566	320 356	181 321 496	23,7908	8,2719	0,00177	6,3386	56,6	177,8	2516,1	566
567	321 489	182 284 263	23,8118	8,2768	0,00176	6,3404	56,7	178,1	2525,0	567
568	322 624	183 250 432	23,8328	8,2816	0,00176	6,3421	56,8	178,4	2533,9	568
569	323 761	184 220 009	23,8537	8,2865	0,00176	6,3439	56,9	178,8	2542,8	569
570	324 900	185 193 000	23,8747	8,2913	0,00175	6,3456	57,0	179,1	2551,8	570
571	326 041	186 169 411	23,8956	8,2962	0,00175	6,3474	57,1	179,4	2560,7	571
572	327 184	187 149 248	23,9165	8,3010	0,00175	6,3491	57,2	179,7	2569,7	572
573	328 329	188 132 517	23,9374	8,3059	0,00175	6,3509	57,3	180,0	2578,7	573

574	329 476	189 119 224	23,9583	8,3107	0,00174	6,3526	57,4	180,3	2587,7	574
575	330 625	190 109 375	23,9792	8,3155	0,00174	6,3544	57,5	180,6	2596,7	575
576	331 776	191 102 976	24,0000	8,3203	0,00174	6,3561	57,6	181,0	2605,8	576
577	332 929	192 100 033	24,0208	8,3251	0,00173	6,3578	57,7	181,3	2614,8	577
578	334 084	193 100 552	24,0416	8,3300	0,00173	6,3596	57,8	181,6	2623,9	578
579	335 241	194 104 539	24,0624	8,3348	0,00173	6,3613	57,9	181,9	2633,0	579
580	336 400	195 112 000	24,0832	8,3396	0,00172	6,3630	58,0	182,2	2642,1	580
581	337 561	196 122 941	24,1039	8,3443	0,00172	6,3647	58,1	182,5	2651,2	581
582	338 724	197 137 368	24,1247	8,3491	0,00172	6,3665	58,2	182,8	2660,3	582
583	339 889	198 155 287	24,1454	8,3539	0,00172	6,3682	58,3	183,2	2669,5	583
584	341 056	199 176 704	24,1661	8,3587	0,00171	6,3699	58,4	183,5	2678,6	584
585	342 225	200 201 625	24,1868	8,3634	0,00171	6,3716	58,5	183,8	2687,8	585
586	343 396	201 230 056	24,2074	8,3682	0,00171	6,3733	58,6	184,1	2697,0	586
587	344 569	202 262 003	24,2281	8,3730	0,00170	6,3750	58,7	184,4	2706,2	587
588	345 744	203 297 472	24,2487	8,3777	0,00170	6,3767	58,8	184,7	2715,5	588
589	346 921	204 336 469	24,2693	8,3825	0,00170	6,3784	58,9	185,0	2724,7	589
590	348 100	205 379 000	24,2899	8,3872	0,00169	6,3801	59,0	185,4	2734,0	590
591	349 281	206 425 071	24,3105	8,3919	0,00169	6,3818	59,1	185,7	2743,2	591
592	350 464	207 474 688	24,3311	8,3967	0,00169	6,3835	59,2	186,0	2752,5	592
593	351 649	208 527 857	24,3516	8,4014	0,00169	6,3852	59,3	186,3	2761,8	593
594	352 836	209 584 584	24,3721	8,4061	0,00168	6,3869	59,4	186,6	2771,2	594
595	354 025	210 644 875	24,3926	8,4108	0,00168	6,3886	59,5	186,9	2780,5	595
596	355 216	211 708 736	24,4131	8,4155	0,00168	6,3902	59,6	187,2	2789,9	596
597	356 409	212 776 173	24,4336	8,4202	0,00168	6,3919	59,7	187,6	2799,2	597
598	357 604	213 847 192	24,4540	8,4249	0,00167	6,3936	59,8	187,9	2808,6	598
599	358 801	214 921 799	24,4745	8,4296	0,00167	6,3953	59,9	188,2	2818,0	599
600	360 000	216 000 000	24,4949	8,4343	0,00167	6,3969	60,0	188,5	2827,4	600

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat}\}n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
601	361 201	217 081 801	24,5153	8,4390	0,00166	6,3986	60,1	188,8	2836,9	601
602	362 404	218 167 208	24,5357	8,4437	0,00166	6,4003	60,2	189,1	2846,3	602
603	363 609	219 256 227	24,5561	8,4484	0,00166	6,4019	60,3	189,4	2855,8	603
604	364 816	220 348 864	24,5764	8,4530	0,00166	6,4036	60,4	189,8	2865,3	604
605	366 025	221 445 125	24,5967	8,4577	0,00165	6,4052	60,5	190,1	2874,8	605
606	367 236	222 545 016	24,6171	8,4623	0,00165	6,4069	60,6	190,4	2884,3	606
607	368 449	223 648 543	24,6374	8,4670	0,00165	6,4085	60,7	190,7	2893,8	607
608	369 664	224 755 712	24,6577	8,4716	0,00164	6,4102	60,8	191,0	2903,3	608
609	370 881	225 866 529	24,6779	8,4763	0,00164	6,4118	60,9	191,3	2912,9	609
610	372 100	226 981 000	24,6982	8,4809	0,00164	6,4135	61,0	191,6	2922,5	**610**
611	373 321	228 099 131	24,7184	8,4856	0,00164	6,4151	61,1	192,0	2932,1	611
612	374 544	229 220 928	24,7386	8,4902	0,00163	6,4167	61,2	192,3	2941,7	612
613	375 769	230 346 397	24,7588	8,4948	0,00163	6,4184	61,3	192,6	2951,3	613
614	376 996	231 475 544	24,7790	8,4994	0,00163	6,4200	61,4	192,9	2960,9	614
615	378 225	232 608 375	24,7992	8,5040	0,00163	6,4216	61,5	193,2	2970,6	615
616	379 456	233 744 896	24,8193	8,5086	0,00162	6,4232	61,6	193,5	2980,2	616
617	380 689	234 885 113	24,8395	8,5132	0,00162	6,4249	61,7	193,8	2989,9	617
618	381 924	236 029 032	24,8596	8,5178	0,00162	6,4265	61,8	194,2	2999,6	618
619	383 161	237 176 659	24,8797	8,5224	0,00162	6,4281	61,9	194,5	3009,3	619
620	384 400	238 328 000	24,8998	8,5270	0,00161	6,4297	62,0	194,8	3019,1	**620**
621	385 641	239 483 061	24,9199	8,5316	0,00161	6,4313	62,1	195,1	3028,8	621
622	386 884	240 641 848	24,9399	8,5362	0,00161	6,4329	62,2	195,4	3038,6	622
623	388 129	241 804 367	24,9600	8,5408	0,00161	6,4345	62,3	195,7	3048,4	623

624	389 376	242 970 624	24,9800	8,5453	0,00160	6,4361	62,4	196,0	3058,2	624
625	390 625	244 140 625	25,0000	8,5499	0,00160	6,4377	62,5	196,3	3068,0	625
626	391 876	245 314 376	25,0200	8,5544	0,00160	6,4393	62,6	196,7	3077,8	626
627	393 129	246 491 883	25,0400	8,5590	0,00159	6,4409	62,7	197,0	3087,6	627
628	394 384	247 673 152	25,0599	8,5635	0,00159	6,4425	62,8	197,3	3097,5	628
629	395 641	248 858 189	25,0799	8,5681	0,00159	6,4441	62,9	197,6	3107,7	629
630	396 900	250 047 000	25,0998	8,5726	0,00159	6,4457	63,0	197,9	3117,2	**630**
631	398 161	251 239 591	25,1197	8,5772	0,00158	6,4473	63,1	198,2	3127,1	631
632	399 424	252 435 968	25,1396	8,5817	0,00158	6,4489	63,2	198,5	3137,1	632
633	400 689	253 636 137	25,1595	8,5862	0,00158	6,4505	63,3	198,9	3147,0	633
634	401 956	254 840 104	25,1794	8,5907	0,00158	6,4521	63,4	199,2	3157,0	634
635	403 225	256 047 875	25,1992	8,5952	0,00157	6,4536	63,5	199,5	3166,9	635
636	404 496	257 259 456	25,2190	8,5997	0,00157	6,4552	63,6	199,8	3176,9	636
637	405 769	258 474 853	25,2389	8,6043	0,00157	6,4568	63,7	200,1	3186,9	637
638	407 044	259 694 072	25,2587	8,6088	0,00157	6,4583	63,8	200,4	3196,9	638
639	408 321	260 917 119	25,2784	8,6132	0,00156	6,4599	63,9	200,7	3206,9	639
640	409 600	262 144 000	25,2982	8,6177	0,00156	6,4615	64,0	201,1	3217,0	**640**
641	410 881	263 374 721	25,3180	8,6222	0,00156	6,4630	64,1	201,4	3227,1	641
642	412 164	264 609 288	25,3377	8,6267	0,00156	6,4646	64,2	201,7	3237,1	642
643	413 449	265 847 707	25,3574	8,6312	0,00156	6,4661	64,3	202,0	3247,2	643
644	414 736	267 089 984	25,3772	8,6357	0,00155	6,4677	64,4	202,3	3257,3	644
645	416 025	268 336 125	25,3969	8,6401	0,00155	6,4693	64,5	202,6	3267,5	645
646	417 316	269 586 136	25,4165	8,6446	0,00155	6,4708	64,6	202,9	3277,6	646
647	418 609	270 840 023	25,4362	8,6490	0,00155	6,4724	64,7	203,3	3287,7	647
648	419 904	272 097 792	25,4558	8,6535	0,00154	6,4739	64,8	203,6	3297,9	648
649	421 201	273 359 449	25,4755	8,6579	0,00154	6,4754	64,9	203,9	3308,1	649
650	422 500	274 625 000	25,4951	8,6624	0,00154	6,4770	65,0	204,2	3318,3	**650**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
651	423 801	275 894 451	25,5147	8,6663	0,00154	6,4785	65,1	204,5	3328,5	651
652	425 104	277 167 808	25,5343	8,6713	0,00153	6,4800	65,2	204,8	3338,8	652
653	426 409	278 445 077	25,5539	8,6757	0,00153	6,4816	65,3	205,1	3349,0	653
654	427 716	279 726 264	25,5734	8,6801	0,00153	6,4831	65,4	205,5	3359,3	654
655	429 025	281 011 375	25,5930	8,6845	0,00153	6,4846	65,5	205,8	3369,6	655
656	430 336	282 300 416	25,6125	8,6890	0,00152	6,4862	65,6	206,1	3379,9	656
657	431 649	283 593 393	25,6320	8,6934	0,00152	6,4877	65,7	206,4	3390,2	657
658	432 964	284 890 312	25,6515	8,6978	0,00152	6,4892	65,8	206,7	3400,5	658
659	434 281	286 191 179	25,6710	8,7022	0,00152	6,4907	65,9	207,0	3410,8	659
660	435 600	287 496 000	25,6905	8,7066	0,00152	6,4922	66,0	207,3	3421,2	**660**
661	436 921	288 804 781	25,7099	8,7110	0,00151	6,4937	66,1	207,7	3431,6	661
662	438 244	290 117 528	25,7294	8,7154	0,00151	6,4953	66,2	208,0	3442,0	662
663	439 569	291 434 247	25,7488	8,7198	0,00151	6,4968	66,3	208,3	3452,4	663
664	440 896	292 754 944	25,7682	8,7241	0,00151	6,4983	66,4	208,6	3462,8	664
665	442 225	294 079 625	25,7876	8,7285	0,00150	6,4998	66,5	208,9	3473,2	665
666	443 556	295 408 296	25,8070	8,7329	0,00150	6,5013	66,6	209,2	3483,7	666
667	444 889	296 740 963	25,8263	8,7373	0,00150	6,5028	66,7	209,5	3494,2	667
668	446 224	298 077 632	25,8457	8,7416	0,00150	6,5043	66,8	209,9	3504,6	668
669	447 561	299 418 309	25,8650	8,7460	0,00149	6,5058	66,9	210,2	3515,1	669
670	448 900	300 763 000	25,8844	8,7503	0,00149	6,5073	67,0	210,5	3525,7	**670**
671	450 241	302 111 711	25,9037	8,7547	0,00149	6,5088	67,1	210,8	3536,2	671
672	451 584	303 464 448	25,9230	8,7590	0,00149	6,5103	67,2	211,1	3546,7	672
673	452 929	304 821 217	25,9422	8,7634	0,00149	6,5118	67,3	211,4	3557,3	673

674	454 276	306 182 024	25,9615	8,7677	0,00148	6,5132	67,4	211,7	3567,9	674
675	455 625	307 546 875	25,9808	8,7721	0,00148	6,5147	67,5	212,1	3578,5	675
676	456 976	308 915 776	26,0000	8,7764	0,00148	6,5162	67,6	212,4	3589,1	676
677	458 329	310 288 733	26,0192	8,7807	0,00148	6,5177	67,7	212,7	3599,7	677
678	459 684	311 665 752	26,0384	8,7850	0,00147	6,5191	67,8	213,0	3610,3	678
679	461 041	313 046 839	26,0576	8,7893	0,00147	6,5206	67,9	213,3	3621,0	679
680	462 400	314 432 000	26,0768	8,7937	0,00147	6,5221	68,0	213,6	3631,7	**680**
681	463 761	315 821 241	26,0960	8,7980	0,00147	6,5236	68,1	213,9	3642,4	681
682	465 124	317 214 568	26,1151	8,8023	0,00147	6,5250	68,2	214,3	3653,1	682
683	466 489	318 611 987	26,1343	8,8066	0,00146	6,5265	68,3	214,6	3663,8	683
684	467 856	320 013 504	26,1534	8,8109	0,00146	6,5280	68,4	214,9	3674,5	684
685	469 225	321 419 125	26,1725	8,8152	0,00146	6,5294	68,5	215,2	3685,3	685
686	470 596	322 828 856	26,1916	8,8194	0,00146	6,5309	68,6	215,5	3696,1	686
687	471 969	324 242 703	26,2107	8,8237	0,00146	6,5323	68,7	215,8	3706,8	687
688	473 344	325 660 672	26,2298	8,8280	0,00145	6,5338	68,8	216,1	3717,6	688
689	474 721	327 082 769	26,2488	8,8323	0,00145	6,5352	68,9	216,5	3728,5	689
690	476 100	328 509 000	26,2679	8,8366	0,00145	6,5367	69,0	216,8	3739,3	**690**
691	477 481	329 939 371	26,2869	8,8408	0,00145	6,5381	69,1	217,1	3750,1	691
692	478 864	331 373 888	26,3059	8,8451	0,00145	6,5396	69,2	217,4	3761,0	692
693	480 249	332 812 557	26,3249	8,8493	0,00144	6,5410	69,3	217,7	3771,9	693
694	481 636	334 255 384	26,3439	8,8536	0,00144	6,5425	69,4	218,0	3782,8	694
695	483 025	335 702 375	26,3629	8,8578	0,00144	6,5439	69,5	218,3	3793,7	695
696	484 416	337 153 536	26,3818	8,8621	0,00144	6,5453	69,6	218,7	3804,6	696
697	485 809	338 608 873	26,4008	8,8663	0,00143	6,5468	69,7	219,0	3815,5	697
698	487 204	340 068 392	26,4197	8,8706	0,00143	6,5482	69,8	219,3	3826,5	698
699	488 601	341 532 099	26,4386	8,8748	0,00143	6,5496	69,9	219,6	3837,5	699
700	490 000	343 000 000	26,4575	8,8790	0,00143	6,5511	70,0	219,9	3848,5	**700**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
701	491 401	344 472 101	26,4764	8,8833	0,00143	6,5525	70,1	220,2	3859,5	701
702	492 804	345 948 408	26,4953	8,8875	0,00142	6,5539	70,2	220,5	3870,5	702
703	494 200	347 428 927	26,5141	8,8917	0,00142	6,5554	70,3	220,9	3881,5	703
704	495 616	348 913 664	26,5330	8,8959	0,00142	6,5568	70,4	221,2	3892,6	704
705	497 025	350 402 625	26,5518	8,9001	0,00142	6,5582	70,5	221,5	3903,6	705
706	498 436	351 895 816	26,5707	8,9043	0,00142	6,5596	70,6	221,8	3914,7	706
707	499 849	353 393 243	26,5895	8,9085	0,00141	6,5610	70,7	222,1	3925,8	707
708	501 264	354 894 912	26,6083	8,9127	0,00141	6,5624	70,8	222,4	3936,9	708
709	502 681	356 400 829	26,6271	8,9169	0,00141	6,5639	70,9	222,7	3948,0	709
710	504 100	357 911 000	26,6458	8,9211	0,00141	6,5653	71,0	223,1	3959,2	710
711	505 521	359 425 431	26,6646	8,9253	0,00141	6,5667	71,1	223,4	3970,4	711
712	506 944	360 944 128	26,6833	8,9295	0,00140	6,5681	71,2	223,7	3981,5	712
713	508 369	362 467 097	26,7021	8,9337	0,00140	6,5695	71,3	224,0	3992,7	713
714	509 796	363 994 344	26,7208	8,9378	0,00140	6,5709	71,4	224,3	4003,9	714
715	511 225	365 525 875	26,7395	8,9420	0,00140	6,5723	71,5	224,6	4015,2	715
716	512 656	367 061 696	26,7582	8,9462	0,00140	6,5737	71,6	224,9	4026,4	716
717	514 089	368 601 813	26,7769	8,9503	0,00139	6,5751	71,7	225,3	4037,6	717
718	515 524	370 146 232	26,7955	8,9545	0,00139	6,5765	71,8	225,6	4048,9	718
719	516 961	371 694 959	26,8142	8,9587	0,00139	6,5779	71,9	225,9	4060,2	719
720	518 400	373 248 000	26,8328	8,9628	0,00139	6,5793	72,0	226,2	4071,5	720
721	519 841	374 805 361	26,8514	8,9670	0,00139	6,5806	72,1	226,5	4082,8	721
722	521 284	376 367 048	26,8701	8,9711	0,00139	6,5820	72,2	226,8	4094,2	722
723	522 729	377 933 067	26,8887	8,9752	0,00138	6,5834	72,3	227,1	4105,5	723

724	524 176	379 503 424	26,9072	8,9794	0,00138	6,5848	72,4	227,5	4116,9	724
725	525 625	381 078 125	26,9258	8,9835	0,00138	6,5862	72,5	227,8	4128,2	725
726	527 076	382 657 176	26,9444	8,9876	0,00138	6,5876	72,6	228,1	4139,6	726
727	528 529	384 240 583	26,9629	8,9918	0,00138	6,5889	72,7	228,4	4151,1	727
728	529 984	385 828 352	26,9815	8,9959	0,00137	6,5903	72,8	228,7	4162,5	728
729	531 441	387 420 489	27,0000	9,0000	0,00137	6,5917	72,9	229,0	4173,9	729
730	532 900	389 017 000	27,0185	9,0041	0,00137	6,5930	73,0	229,3	4185,4	730
731	534 361	390 617 891	27,0370	9,0082	0,00137	6,5944	73,1	229,7	4196,9	731
732	535 824	392 223 168	27,0555	9,0123	0,00137	6,5958	73,2	230,0	4208,4	732
733	537 289	393 832 837	27,0740	9,0164	0,00136	6,5971	73,3	230,3	4219,9	733
734	538 756	395 446 904	27,0924	9,0205	0,00136	6,5985	73,4	230,6	4231,4	734
735	540 225	397 065 375	27,1109	9,0246	0,00136	6,5999	73,5	230,9	4242,9	735
736	541 696	398 688 256	27,1293	9,0287	0,00136	6,6012	73,6	231,2	4254,5	736
737	543 169	400 315 553	27,1477	9,0328	0,00136	6,6026	73,7	231,5	4266,0	737
738	544 644	401 947 272	27,1662	9,0369	0,00136	6,6039	73,8	231,8	4277,6	738
739	546 121	403 583 419	27,1846	9,0410	0,00135	6,6053	73,9	232,2	4289,2	739
740	547 600	405 224 000	27,2029	9,0450	0,00135	6,6066	74,0	232,5	4300,3	740
741	549 081	406 869 021	27,2213	9,0491	0,00135	6,6080	74,1	232,8	4312,5	741
742	550 564	408 518 488	27,2397	9,0532	0,00135	6,6093	74,2	233,1	4324,1	742
743	552 049	410 172 407	27,2580	9,0572	0,00135	6,6107	74,3	233,4	4335,8	743
744	553 536	411 830 784	27,2764	9,0613	0,00134	6,6120	74,4	233,7	4347,5	744
745	555 025	413 493 625	27,2947	9,0654	0,00134	6,6134	74,5	234,0	4359,2	745
746	556 516	415 160 936	27,3130	9,0694	0,00134	6,6147	74,6	234,4	4370,9	746
747	558 009	416 832 723	27,3313	9,0735	0,00134	6,6161	74,7	234,7	4382,6	747
748	559 504	418 508 992	27,3496	9,0775	0,00134	6,6174	74,8	235,0	4394,3	748
749	561 001	420 189 749	27,3679	9,0816	0,00134	6,6187	74,9	235,3	4406,1	749
750	562 500	421 875 000	27,3861	9,0856	0,00133	6,6201	75,0	235,6	4417,6	750

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\left.\begin{matrix}\log\\ \text{nat}\end{matrix}\right\}n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
751	564 001	423 564 751	27,4044	9,0896	0,00133	6,6214	75,1	235,9	4429,7	751
752	565 504	425 259 008	27,4226	9,0937	0,00133	6,6227	75,2	236,2	4441,5	752
753	567 009	426 957 777	27,4408	9,0977	0,00133	6,6241	75,3	236,6	4453,3	753
754	568 516	428 661 064	27,4591	9,1017	0 00133	6,6254	75,4	236,9	4465,1	754
755	570 025	430 368 875	27,4773	9,1057	0,00132	6,6267	75,5	237,2	4477,0	755
756	571 536	432 081 216	27,4955	9,1098	0 00132	6,6280	75,6	237,5	4488,8	756
757	573 049	433 798 093	27,5136	9,1138	0,00132	6,6294	75,7	237,8	4500,7	757
758	574 564	435 519 512	27,5318	9,1178	0,00132	6,6307	75,8	238,1	4512,6	758
759	576 081	437 245 479	27,5500	9,1218	0,00132	6,6320	75,9	238,4	4524,5	759
760	577 600	438 976 000	27,5681	9,1258	0,00132	6,6333	76,0	238,8	4536,5	**760**
761	579 121	440 711 081	27,5862	9,1298	0,00131	6,6346	76,1	239,1	4548,4	761
762	580 644	442 450 728	27,6043	9,1338	0,00131	6,6359	76,2	239,4	4560,4	762
763	582 169	444 194 947	27,6225	9,1378	0,00131	6,6373	76,3	239,7	4572,3	763
764	583 696	445 943 744	27,6405	9,1418	0,00131	6,6386	76,4	240,0	4584,3	764
765	585 225	447 697 125	27,6586	9,1458	0,00131	6,6399	76,5	240,3	4596,3	765
766	586 756	449 455 096	27,6767	9,1498	0,00131	6,6412	76,6	240,6	4608,4	766
767	588 289	451 217 663	27,6948	9,1537	0,00130	6,6425	76,7	241,0	4620,4	767
768	589 824	452 984 832	27,7128	9,1577	0,00130	6,6438	76,8	241,3	4632,5	768
769	591 361	454 756 609	27,7308	9,1617	0,00130	6,6451	76,9	241,6	4644,5	769
770	592 900	456 533 000	27,7489	9,1657	0,00130	6,6464	77,0	241,9	4656,6	**770**
771	594 441	458 314 011	27,7669	9,1696	0,00130	6,6477	77,1	242,2	4668,7	771
772	595 984	460 099 648	27,7849	9,1736	0,00130	6,6490	77,2	242,5	4680,8	772
773	597 529	461 889 917	27,8029	9,1775	0,00129	6,6503	77,3	242,8	4698,0	773

774	599 076	463 684 824	27,8209	9,1815	0,00129	6,6516	77,4	243,2	4705,1	774
775	600 625	465 484 375	27,8388	9,1855	0,00129	6,6529	77,5	243,5	4717,3	775
776	602 176	467 288 576	27,8568	9,1894	0,00129	6,6542	77,6	243,8	4729,5	776
777	603 729	469 097 433	27,8747	9,1933	0,00129	6,6554	77,7	244,1	4741,7	777
778	605 284	470 910 952	27,8927	9,1973	0,00129	6,6567	77,8	244,4	4753,9	778
779	606 841	472 729 139	27,9106	9,2012	0,00128	6,6580	77,9	244,7	4766,1	779
780	608 400	474 552 000	27,9285	9,2052	0,00128	6,6593	78,0	245,0	4778,4	780
781	609 961	476 379 541	27,9464	9,2091	0,00128	6,6606	78,1	245,4	4790,6	781
782	611 524	478 211 768	27,9643	9,2130	0,00128	6,6619	78,2	245,7	4802,9	782
783	613 089	480 048 687	27,9821	9,2170	0,00128	6,6631	78,3	246,0	4815,2	783
784	614 656	481 890 304	28,0000	9,2209	0,00128	6,6644	78,4	246,3	4827,5	784
785	616 225	483 736 625	28,0179	9,2248	0,00127	6,6657	78,5	246,6	4839,8	785
786	617 796	485 587 656	28,0357	9,2287	0,00127	6,6670	78,6	246,9	4852,2	786
787	619 369	487 443 403	28,0535	9,2326	0,00127	6,6682	78,7	247,2	4864,5	787
788	620 944	489 303 872	28,0713	9,2365	0,00127	6,6695	78,8	247,6	4876,9	788
789	622 521	491 169 069	28,0891	9,2404	0,00127	6,6708	78,9	247,9	4889,3	789
790	624 100	493 039 000	28,1069	9,2443	0,00127	6,6720	79,0	248,2	4901,7	790
791	625 681	494 913 671	28,1247	9,2482	0,00126	6,6733	79,1	248,5	4914,1	791
792	627 264	496 793 088	28,1425	9,2521	0,00126	6,6746	79,2	248,8	4926,5	792
793	628 849	498 677 257	28,1603	9,2560	0,00126	6,6758	79,3	249,1	4939,0	793
794	630 436	500 566 184	28,1780	9,2599	0,00126	6,6771	79,4	249,4	4951,4	794
795	632 025	502 459 875	28,1957	9,2638	0,00126	6,6783	79,5	249,8	4963,9	795
796	633 616	504 358 336	28,2135	9,2677	0,00126	6,6796	79,6	250,1	4976,4	796
797	635 209	506 261 573	28,2312	9,2716	0,00125	6,6809	79,7	250,4	4988,9	797
798	636 804	508 169 592	28,2489	9,2754	0,00125	6,6821	79,8	250,7	5001,4	798
799	638 401	510 082 399	28,2666	9,2793	0,00125	6,6834	79,9	251,0	5014,0	799
800	640 000	512 000 000	28,2843	9,2832	0,00125	6,6846	80,0	251,3	5026,5	800

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\left.\begin{matrix}\log\\ \text{nat}\end{matrix}\right\}n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
801	641 601	513 922 401	28,3019	9,2870	0,00125	6,6859	80,1	251,6	5039,1	801
802	643 204	515 849 608	28,3196	9,2909	0,00125	6,6871	80,2	252,0	5051,7	802
803	644 809	517 781 627	28,3373	9,2948	0,00125	6,6883	80,3	252,3	5064,3	803
804	646 416	519 718 464	28,3549	9,2986	0,00124	6,6896	80,4	252,6	5076,9	804
805	648 025	521 660 125	28,3725	9,3025	0,00124	6,6908	80,5	252,9	5089,6	805
806	649 636	523 606 616	28,3901	9,3063	0,00124	6,6921	80,6	253,2	5102,2	806
807	651 249	525 557 943	28,4077	9,3102	0,00124	6,6933	80,7	253,5	5114,9	807
808	652 864	527 514 112	28,4253	9,3140	0,00124	6,6946	80,8	253,8	5127,6	808
809	654 481	529 475 129	28,4429	9,3179	0,00124	6,6958	80,9	254,2	5140,3	809
810	656 100	531 441 000	28,4605	9,3217	0,00123	6,6970	81,0	254,5	5153,0	810
811	657 721	533 411 731	28,4781	9,3255	0,00123	6,6983	81,1	254,8	5165,7	811
812	659 344	535 387 328	28,4956	9,3294	0,00123	6,6995	81,2	255,1	5178,5	812
813	660 969	537 367 797	28,5132	9,3332	0,00123	6,7007	81,3	255,4	5191,2	813
814	662 596	539 353 144	28,5307	9,3370	0,00123	6,7020	81,4	255,7	5204,0	814
815	664 225	541 343 375	28,5482	9,3408	0,00123	6,7032	81,5	256,0	5216,8	815
816	665 856	543 338 496	28,5657	9,3447	0,00123	6,7044	81,6	256,4	5229,6	816
817	667 489	545 338 513	28,5832	9,3485	0,00122	6,7056	81,7	256,7	5242,4	817
818	669 124	547 343 432	28,6007	9,3523	0,00122	6,7069	81,8	257,0	5255,3	818
819	670 761	549 353 259	28,6182	9,3561	0,00122	6,7081	81,9	257,3	5268,1	819
820	672 400	551 368 000	28,6356	9,3599	0,00122	6,7093	82,0	257,6	5281,0	820
821	674 041	553 387 661	28,6531	9,3637	0,00122	6,7105	82,1	257,9	5293,9	821
822	675 684	555 412 248	28,6705	9,3675	0,00122	6,7117	82,2	258,2	5306,8	822
823	677 329	557 441 767	28,6880	9,3713	0,00122	6,7130	82,3	258,6	5319,7	823

824	678 976	559 476 224	28,7054	9,3751	0,00121	6,7142	82,4	258,9	5332,7	824
825	680 625	561 515 625	28,7228	9,3789	0,00121	6,7154	82,5	259,2	5345,6	825
826	682 276	563 559 976	28,7402	9,3827	0,00121	6,7166	82,6	259,5	5358,6	826
827	683 929	565 609 283	28,7576	9,3865	0,00121	6,7178	82,7	259,8	5371,6	827
828	685 584	567 663 552	28,7750	9,3902	0,00121	6,7190	82,8	260,1	5384,6	828
829	687 241	569 722 789	28,7924	9,3940	0,00121	6,7202	82,9	260,4	5397,6	829
830	688 900	571 787 000	28,8097	9,3978	0,00120	6,7214	83,0	260,8	5410,6	830
831	690 561	573 856 191	28,8271	9,4016	0,00120	6,7226	83,1	261,1	5423,7	831
832	692 224	575 930 368	28,8444	9,4053	0,00120	6,7238	83,2	261,4	5436,7	832
833	693 889	578 009 537	28,8617	9,4091	0,00120	6,7250	83,3	261,7	5449,8	833
834	695 556	580 093 704	28,8791	9,4129	0,00120	6,7262	83,4	262,0	5462,9	834
835	697 225	582 182 875	28,8964	9,4166	0,00120	6,7274	83,5	262,3	5476,0	835
836	698 896	584 277 056	28,9137	9,4204	0,00120	6,7286	83,6	262,6	5489,1	836
837	700 569	586 376 253	28,9310	9,4241	0,00119	6,7298	83,7	263,0	5502,3	837
838	702 244	588 480 472	28,9482	9,4279	0,00119	6,7310	83,8	263,3	5515,4	838
839	703 921	590 589 719	28,9655	9,4316	0,00119	6,7322	83,9	263,6	5528,6	839
840	705 600	592 704 000	28,9828	9,4354	0,00119	6,7334	84,0	263,9	5541,8	840
841	707 281	594 823 321	29,0000	9,4391	0,00119	6,7346	84,1	264,2	5555,0	841
842	708 964	596 947 688	29,0172	9,4429	0,00119	6,7358	84,2	264,5	5568,2	842
843	710 649	599 077 107	29,0345	9,4466	0,00119	6,7370	84,3	264,8	5581,4	843
844	712 336	601 211 584	29,0517	9,4503	0,00118	6,7382	84,4	265,2	5594,7	844
845	714 025	603 351 125	29,0689	9,4541	0,00118	6,7393	84,5	265,5	5607,9	845
846	715 716	605 495 736	29,0861	9,4578	0,00118	6,7405	84,6	265,8	5621,2	846
847	717 409	607 645 423	29,1033	9,4615	0,00118	6,7417	84,7	266,1	5634,5	847
848	719 104	609 800 192	29,1204	9,4652	0,00118	6,7429	84,8	266,4	5647,8	848
849	720 801	611 960 049	29,1376	9,4690	0,00118	6,7441	84,9	266,7	5661,2	849
850	722 500	614 125 000	29,1548	9,4727	0,00118	6,7452	85,0	267,0	5674,5	850

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
851	724 201	616 295 051	29,1719	9,4764	0,00118	6,7464	85,1	267,3	5687,9	851
852	725 904	618 470 208	29,1890	9,4801	0,00117	6,7476	85,2	267,7	5701,2	852
853	727 609	620 650 477	29,2062	9,4838	0,00117	6,7488	85,3	268,0	5714,6	853
854	729 316	622 835 864	29,2233	9,4875	0,00117	6,7499	85,4	268,3	5728,0	854
855	731 025	625 026 375	29,2404	9,4912	0,00117	6,7511	85,5	268,6	5741,5	855
856	732 736	627 222 016	29,2575	9,4949	0,00117	6,7523	85,6	268,9	5754,9	856
857	734 449	629 422 793	29,2746	9,4986	0,00117	6,7534	85,7	269,2	5768,3	857
858	736 164	631 628 712	29,2916	9,5023	0,00117	6,7546	85,8	269,5	5781,8	858
859	737 881	633 839 779	29,3087	9,5060	0,00116	6,7558	85,9	269,9	5795,3	859
860	739 600	636 056 000	29,3258	9,5097	0,00116	6,7569	86,0	270,2	5808,8	**860**
861	741 321	638 277 381	29,3428	9,5134	0,00116	6,7581	86,1	270,5	5822,3	861
862	743 044	640 503 928	29,3598	9,5171	0,00116	6,7593	86,2	270,8	5835,9	862
863	744 769	642 735 647	29,3769	9,5207	0,00116	6,7604	86,3	271,1	5849,4	863
864	746 496	644 972 544	29,3939	9,5244	0,00116	6,7616	86,4	271,4	5863,0	864
865	748 225	647 214 625	29,4109	9,5281	0,00116	6,7627	86,5	271,7	5876,5	865
866	749 956	649 461 896	29,4279	9,5317	0,00115	6,7639	86,6	272,1	5890,1	866
867	751 689	651 714 363	29,4449	9,5354	0,00115	6,7650	86,7	272,4	5903,8	867
868	753 424	653 972 032	29,4618	9,5391	0,00115	6,7662	86,8	272,7	5917,4	868
869	755 161	656 284 909	29,4788	9,5427	0,00115	6,7673	86,9	273,0	5931,0	869
870	756 900	658 503 000	29,4958	9,5464	0,00115	6,7685	87,0	273,3	5944,7	**870**
871	758 641	660 776 311	29,5127	9,5501	0,00115	6,7696	87,1	273,6	5958,4	871
872	760 384	663 054 848	29,5296	9,5537	0,00115	6,7708	87,2	273,9	5972,0	872
873	762 129	665 338 617	29,5466	9,5574	0,00115	6,7719	87,3	274,3	5985,7	873

874	763 876	667 627 624	29,5685	9,5610	0,00114	6,7731	87,4	274,6	5999,5	874
875	765 625	669 921 875	29,5804	9,5647	0,00114	6,7742	87,5	274,9	6013,2	875
876	767 376	672 221 376	29,5973	9,5683	0,00114	6,7754	87,6	275,2	6027,0	876
877	769 129	674 526 133	29,6142	9,5719	0,00114	6,7765	87,7	275,5	6040,7	877
878	770 884	676 836 152	29,6311	9,5756	0,00114	6,7776	87,8	275,8	6054,5	878
879	772 641	679 151 439	29,6479	9,5792	0,00114	6,7788	87,9	276,1	6068.3	879
880	774 400	681 472 000	29,6648	9,5828	0,00114	6,7799	88,0	276,5	6082,1	880
881	776 161	683 797 841	29,6816	9,5865	0,00114	6,7811	88,1	276,8	6096,0	881
882	777 924	686 128 968	29,6985	9,5901	0,00113	6,7822	88,2	277,1	6109,8	882
883	779 689	688 465 387	29,7153	9,5937	0,00113	6,7833	88,3	277,4	6123,7	883
884	781 456	690 807 104	29,7321	9,5973	0,00113	6,7845	88,4	277,7	6137,5	884
885	783 225	693 154 125	29,7489	9,6010	0,00113	6,7856	88,5	278,0	6151,4	885
886	784 996	695 506 456	29,7658	9,6046	0,00113	6,7867	88,6	278,3	6165,3	886
887	786 769	697 864 103	29,7825	9,6082	0,00113	6,7878	88,7	278,7	6179,3	887
888	788 544	700 227 072	29,7993	9,6118	0,00113	6,7890	88,8	279,0	6193,2	888
889	790 321	702 595 369	29,8161	9,6154	0,00112	6,7901	88,9	279,3	6207,2	889
890	792 100	704 969 000	29,8329	9,6190	0,00112	6,7912	89,0	279,6	6221,1	890
891	793 881	707 347 971	29,8496	9,6226	0,00112	6,7923	89,1	279,9	6235,1	891
892	795 664	709 732 288	29,8664	9,6262	0,00112	6,7935	89,2	280,2	6249,1	892
893	797 449	712 121 957	29,8831	9,6298	0,00112	6,7946	89,3	280,5	6263,1	893
894	799 236	714 516 984	29,8998	9,6334	0,00112	6,7957	89,4	280,9	6577,2	894
895	801 025	716 917 375	29,9166	9,6370	0,70112	6,7968	89,5	281,2	6291,2	895
896	802 816	719 323 136	29,9333	9,6406	0,00112	6,7979	89,6	281,5	6305,3	896
897	804 609	721 734 273	29,9500	9,6442	0,00111	6,7991	89,7	281,8	6319,4	897
898	806 404	724 150 792	29,9666	9,6477	0,00111	6,8002	89,8	282,1	6333,5	898
899	808 201	726 572 699	29,9833	9,6513	0,00111	6,8013	89,9	282,4	6347,6	899
900	810 000	729 000 000	30,0000	9,6549	0,00111	6,8024	90,0	282,7	6361,7	900

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
901	811 801	731 432 701	30,0167	9,6585	0,00111	6,8085	90,1	283,1	6375,9	901
902	813 604	733 870 808	30,0333	9,6620	0,00111	6,8046	90,2	283,4	6390,0	902
903	815 409	736 314 327	30,0500	9,6656	0,00111	6,8057	90,3	283,7	6404,2	903
904	817 216	738 763 264	30,0666	9,6692	0,00111	6,8068	90,4	284,0	6418,4	904
905	819 025	741 217 625	30,0832	9,6727	0,00110	6,8079	90,5	284,3	6432,6	905
906	820 836	743 677 416	30,0998	9,6763	0,00110	6,8090	90,6	284,6	6446,8	906
907	822 649	746 142 643	30,1164	9,6799	0,00110	6,8101	90,7	284,9	6461,1	907
908	824 464	748 613 312	30,1330	9,6834	0,00110	6,8112	90,8	285,3	6475,3	908
909	826 281	751 089 429	30,1496	9,6870	0,00110	6,8123	90,9	285,6	6489,6	909
910	828 100	753 571 000	30,1662	9,6905	0,00110	6,8134	91,0	285,9	6503,9	910
911	829 921	756 058 031	30,1828	9,6941	0,00110	6,8145	91,1	286,2	6518,2	911
912	831 744	758 550 528	30,1993	9,6976	0,00110	6,8156	91,2	286,5	6532,5	912
913	833 569	761 048 497	30,2159	9,7012	0,00110	6,8167	91,3	286,8	6546,8	913
914	835 396	763 551 944	30,2324	9,7047	0,00109	6,8178	91,4	287,1	6561,2	914
915	837 225	766 060 875	30,2490	9,7082	0,00109	6,8189	91,5	287,5	6575,5	915
916	839 056	768 575 296	30,2655	9,7118	0,00109	6,8200	91,6	287,8	6589,8	916
917	840 889	771 095 213	30,2820	9,7153	0,00109	6,8211	91,7	288,1	6604,3	917
918	842 724	773 620 632	30,2985	9,7188	0,00109	6,8222	91,8	288,4	6618,7	918
919	844 561	776 151 559	30,3150	9,7224	0,00109	6,8233	91,9	288,7	6633,2	919
920	846 400	778 688 000	30,3315	9,7259	0,00109	6,8244	92,0	289,0	6647,6	920
921	848 241	781 229 961	30,3480	9,7294	0,00109	6,8255	92,1	289,3	6662,1	921
922	850 084	783 777 448	30,3645	9,7329	0,00108	6,8266	92,2	289,7	6676,5	922
923	851 929	786 330 467	30,3809	9,7364	0,00108	6,8276	92,3	290,0	6691,0	923

924	853 776	788 889 024	30,3974	9,7400	0,00108	6,8287	92,4	290,3	6705,5	924
925	855 625	791 453 125	30,4138	9,7435	0,00108	6,8298	92,5	290,6	6720,1	925
926	857 476	794 022 776	30,4302	9,7470	0,00108	6,8309	92,6	290,9	6734,6	926
927	859 329	796 597 983	30,4467	9,7505	0,00108	6,8319	92,7	291,2	6749,2	927
928	861 184	799 178 752	30,4631	9,7540	0,00108	6,8330	92,8	291,5	6763,7	928
929	863 041	801 765 089	30,4795	9,7575	0,00108	6,8341	92,9	291,9	6778,3	929
930	864 900	804 357 000	30,4959	9,7610	0,00108	6,8352	93,0	292,2	6792,9	**930**
931	866 761	806 954 491	30,5123	9,7645	0,00107	6,8363	93,1	292,5	6807,5	931
932	868 624	809 557 568	30,5287	9,7680	0,00107	6,8373	93,2	292,8	6822,2	932
933	870 489	812 166 237	30,5450	9,7715	0,00107	6,8384	93,3	293,1	6836,8	933
934	872 356	814 780 504	30,5614	9,7750	0,00107	6,8395	93,4	293,4	6851,5	934
935	874 225	817 400 375	30,5778	9,7785	0,00107	6,8406	93,5	293,7	6866,1	935
936	876 096	820 025 856	30,5941	9,7819	0,00107	6,8416	93,6	294,1	6880,8	936
937	877 969	822 656 953	30,6105	9,7854	0,00107	6,8427	93,7	294,4	6895,6	937
938	879 844	825 293 672	30,6268	9,7889	0,00107	6,8438	93,8	294,7	6910,3	938
939	881 721	827 936 019	30,6431	9,7924	0,00106	6,8448	93,9	295,0	6925,0	939
940	883 600	830 584 000	30,6594	9,7959	0,00106	6,8459	94,0	295,3	6939,8	**940**
941	885 481	833 237 621	30,6757	9,7993	0,00106	6,8469	94,1	295,6	6954,6	941
942	887 364	835 896 888	30,6920	9,8028	0,00106	6,8480	94,2	295,9	6969,3	942
943	889 249	838 561 807	30,7083	9,8063	0,00106	6,8491	94,3	296,3	6984,1	943
944	891 136	841 232 384	30,7246	9,8097	0,00106	6,8501	94,4	296,6	6999,0	944
945	893 025	843 908 625	30,7409	9,8132	0,00106	6,8512	94,5	296,9	7013,8	945
946	894 916	846 590 536	30,7571	9,8167	0,00106	6,8522	94,6	297,2	7028,7	946
947	896 809	849 278 123	30,7734	9,8201	0,00106	6,8533	94,7	297,5	7043,5	947
948	898 704	851 971 392	30,7896	9,8236	0,00105	6,8543	94,8	297,8	7058,4	948
949	900 601	854 670 349	30,8058	9,8270	0,00105	6,8554	94,9	298,1	7073,3	949
950	902 500	857 375 000	30,8221	9,8305	0,00105	6,8565	95,0	298,5	7088,2	**950**

N.B.—Decimal points indicated by commas.

n	n^2	n^3	$\sqrt{n}$	$\sqrt[3]{n}$	$\frac{1}{n}$	$\log_{nat} n$	d=0,1 n	πd	$\frac{1}{4}\pi d^2$	n
951	904 401	860 085 351	30,8383	9,8339	0,00105	6,8575	95,1	298,8	7103,1	951
952	906 304	862 801 408	30,8545	9,8374	0,00105	6,8586	95,2	299,1	7118,1	952
953	908 209	865 523 177	30,8707	9,8408	0,00105	6,8596	95,3	299,4	7133,1	953
954	910 116	868 250 664	30,8869	9,8443	0,00105	6,8607	95,4	299,7	7148,0	954
955	912 025	870 983 875	30,9031	9,8477	0,00105	6,8617	95,5	300,0	7163,0	955
956	913 936	873 722 816	30,9192	9,8511	0,00105	6,8628	95,6	300,3	7178,0	956
957	915 849	876 467 493	30,9354	9,8546	0,00104	6,8638	95,7	300,7	7193,1	957
958	917 764	879 217 912	30,9516	9,8580	0,00104	6,8649	95,8	301,0	7208,1	958
959	919 681	881 974 079	30,9677	9,8614	0,00104	6,8659	95,9	301,3	7223,2	959
960	921 600	884 736 000	30,9839	9,8648	0,00104	6,8669	96,0	301,6	7238,2	**960**
961	923 521	887 503 681	31,0000	9,8683	0,00104	6,8680	96,1	301,9	7253,3	961
962	925 444	890 277 128	31,0161	9,8717	0,00104	6,8690	96,2	302,2	7268,4	962
963	927 369	893 056 347	31,0322	9,8751	0,00104	6,8700	96,3	302,5	7283,5	963
964	929 296	895 841 344	31,0483	9,8785	0,00104	6,8711	96,4	302,8	7298,7	964
965	931 225	898 632 125	31,0644	9,8819	0,00104	6,8721	96,5	303,2	7313,8	965
966	933 156	901 428 696	31,0805	9,8854	0,00104	6,8732	96,6	303,5	7329,0	966
967	935 089	904 231 063	31,0966	9,8888	0,00103	6,8742	96,7	303,8	7344,2	967
968	937 024	907 039 232	31,1127	9,8922	0,00103	6,8752	96,8	304,1	7359,4	968
969	938 961	909 853 209	31,1288	9,8956	0,00103	6,8763	96,9	304,4	7374,6	969
970	940 900	912 673 000	31,1448	9,8990	0,00103	6,8773	97,0	304,7	7389,8	**970**
971	942 841	915 498 611	31,1609	9,9024	0,00103	6,8783	97,1	305,0	7405,1	971
972	944 784	918 330 048	31,1769	9,9058	0,00103	6,8794	97,2	305,4	7420,3	972
973	946 729	921 167 317	31,1929	9,9092	0,00103	6,8804	97,3	305,7	7435,6	973

974	948 676	924 010 424	31,2090	9,9126	0,00103	6,8814	97,4	306,0	7450,9	974
975	950 625	926 859 375	31,2250	9,9160	0,00103	6,8824	97,5	306,3	7466,2	975
976	952 576	929 714 176	31,2410	9,9194	0,00102	6,8835	97,6	306,6	7481,5	976
977	954 529	932 574 833	31,2570	9,9227	0,00102	6,8845	97,7	306,9	7496,9	977
978	956 484	935 441 352	31,2730	9,9261	0,00102	6,8855	97,8	307,2	7512,2	978
979	958 441	938 313 739	31,2890	9,9295	0,00102	6,8865	97,9	307,6	7527,6	979
980	960 400	941 192 000	31,3050	9,9329	0,00102	6,8875	98,0	307,9	7543,0	**980**
981	962 361	944 076 141	31,3209	9,9363	0,00102	6,8886	98,1	308,2	7558,4	981
982	964 324	946 966 168	31,3369	9,9396	0,00102	6,8896	98,2	308,5	7573,8	982
983	966 289	949 862 087	31,3528	9,9430	0,00102	6,8906	98,3	308,8	7589,2	983
984	968 256	952 763 904	31,3688	9,9464	0,00102	6,8916	98,4	309,1	7604,7	984
985	970 225	955 671 625	31,3847	9,9497	0,00102	6,8926	98,5	309,4	7620,1	985
986	972 196	958 585 256	31,4006	9,9531	0,00101	6,8937	98,6	309,8	7635,6	986
987	974 169	961 504 803	31,4166	9,9565	0,00101	6,8947	98,7	310,1	7651,1	987
988	976 144	964 430 272	31,4325	9,9598	0,00101	6,8957	98,8	310,4	7666,6	988
989	978 121	967 361 669	31,4484	9,9632	0,00101	6,8967	98,9	310,7	7682,1	989
990	980 100	970 299 000	31,4643	9,9666	0,00101	6,8977	99,0	311,0	7697,7	**990**
991	982 081	973 242 271	31,4802	9,9699	0,00101	6,8987	99,1	311,3	7713,2	991
992	984 064	976 191 488	31,4960	9,9733	0,00101	6,8997	99,2	311,6	7728,8	992
993	986 049	979 146 657	31,5119	9,9766	0,00101	6,9007	99,3	312,0	7744,4	993
994	988 036	982 107 784	31,5278	9,9800	0,00101	6,9017	99,4	312,3	7760,0	994
995	990 025	985 074 875	31,5436	9,9833	0,00101	6,9027	99,5	312,6	7775,6	995
996	992 016	988 047 936	31,5595	9,9866	0,00100	6,9037	99,6	312,9	7791,3	996
997	994 009	991 026 973	31,5753	9,9900	0,00100	6,9048	99,7	313,2	7806,9	997
998	996 004	994 011 992	31,5911	9,9933	0,00100	6,9058	99,8	313,5	7822,6	998
999	998 001	997 002 999	31,6070	9,9967	0,00100	6,9068	99,9	313,8	7838,3	999
1 000	1 000 000	1 000 000 000	31,6228	10,0000	0,00100	6,9078	100,0	314,2	7854,0	**1000**

N.B.—Decimal points indicated by commas.

Table No. II.

Common Logarithms from 1 to 100.

Nr.	0	1	2	3	4	5	6	7	8	9	Difference.
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	43—40
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	39—36
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	36—34
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	34—31
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	31—29
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	29—27
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	27—26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	26—25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24—23
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	23—22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	22—20
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	21—19
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	20—19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19—18
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18—17
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	18—17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	17—16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16—15
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	16—15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15—14
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	15—14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14—13
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	14—13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13—12
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	13—12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12—11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12—11
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	12—11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	12—11

Table No. II.

Common Logarithms from 1 to 100.

Nr.	0	1	2	3	4	5	6	7	8	9	Difference.
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11—10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	11—10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	11—10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10—9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10—9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	10—9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	10—9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9—8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9—8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9—8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	9—8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	9—8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8—7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8—7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8—7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	8—7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	8—7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	8—7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	8—7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7—6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7—6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7—6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	7—6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7—6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	7—6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	7—6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	7—6

Table No. II.

Common Logarithms from 1 to 100.

Nr.	0	1	2	3	4	5	6	7	8	9	Difference.
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	7—6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	7—6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6—5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6—5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6—5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6—5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6—5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6—5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	6—5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	6—5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	6—5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	6—5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	6—5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	6—5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	6—5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5—4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5—4
90	9542	9547	9552	9557	9562	9567	9571	9576	9581	9586	5—4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5—4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5—4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5—4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5—4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5—4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	5—4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	5—4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	5—4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	5—4

Table No. III.—Sines and Cosines.

Degree	Cosines							Degree
	0′	10′	20′	30′	40′	50′	60′	
0	1,0000	1,0000	1,0000	1,0000	0,9999	0,9999	0,9999	89
1	0,9999	0,9998	0,9997	0,9997	0,9996	0,9995	0,9994	88
2	0,9994	0,9993	0,9992	0,9991	0,9989	0,9988	0,9986	87
3	0,9986	0,9985	0,9983	0,9981	0,9980	0,9978	0,9976	86
4	0,9976	0,9974	0,9971	0,9969	0,9967	0,9964	0,9962	85
5	0,9962	0,9959	0,9957	0,9954	0,9951	0,9948	0,9945	84
6	0,9945	0,9942	0,9939	0,9936	0,9932	0,9929	0,9926	83
7	0,9926	0,9922	0,9918	0,9914	0,9911	0,9907	0,9903	82
8	0,9903	0,9899	0,9894	0,9890	0,9886	0,9881	0,9877	81
9	0,9877	0,9872	0,9868	0,9863	0,9858	0,9853	0,9848	80
10	0,9848	0,9843	0,9838	0,9833	0,9827	0,9822	0,9816	79
11	0,9816	0,9811	0,9805	0,9799	0,9793	0,9788	0,9782	78
12	0,9782	0,9775	0,9769	0,9763	0,9757	0,9750	0,9744	77
13	0,9744	0,9737	0,9730	0,9724	0,9717	0,9710	0,9703	76
14	0,9703	0,9696	0,9689	0,9682	0,9674	0,9667	0,9659	75
15	0,9659	0,9652	0,9644	0,9636	0,9629	0,9621	0,9613	74
16	0,9613	0,9605	0,9596	0,9588	0,9580	0,9572	0,9563	73
17	0,9563	0,9555	0,9546	0,9537	0,9528	0,9520	0,9511	72
18	0,9511	0,9502	0,9492	0,9483	0,9474	0,9465	0,9455	71
19	0,9455	0,9446	0,9436	0,9426	0,9417	0,9407	0,9397	70
20	0,9397	0,9387	0,9377	0,9367	0,9357	0,9346	0,9336	69
21	0,9336	0,9325	0,9315	0,9304	0,9294	0,9283	0,9272	68
22	0,9272	0,9261	0,9250	0,9239	0,9228	0,9216	0,9205	67
23	0,9205	0,9194	0,9182	0,9171	0,9159	0,9147	0,9136	66
24	0,9136	0,9124	0,9112	0,9100	0,9088	0,9075	0,9063	65
25	0,9063	0,9051	0,9038	0,9026	0,9013	0,9001	0,8988	64
26	0,8988	0,8975	0,8962	0,8949	0,8936	0,8923	0,8910	63
27	0,8910	0,8897	0,8884	0,8870	0,8857	0,8843	0,8830	62
28	0,8830	0,8816	0,8802	0,8788	0,8774	0,8760	0,8746	61
29	0,8746	0,8732	0,8718	0,8704	0,8689	0,8675	0,8660	60
30	0,8660	0,8646	0,8631	0,8616	0,8602	0,8587	0,8572	59
31	0,8572	0,8557	0,8542	0,8526	0,8511	0,8496	0,8481	58
32	0,8481	0,8465	0,8450	0,8434	0,8418	0,8403	0,8387	57
33	0,8387	0,8371	0,8355	0,8339	0,8323	0,8307	0,8290	56
34	0,8290	0,8274	0,8258	0,8241	0,8225	0,8208	0,8192	55
35	0,8193	0,8175	0,8158	0,8141	0,8124	0,8107	0,8090	54
36	0,8090	0,8073	0,8056	0,8039	0,8021	0,8004	0,7986	53
37	0,7986	0,7969	0,7951	0,7934	0,7916	0,7898	0,7880	52
38	0,7880	0,7862	0,7844	0,7826	0,7808	0,7790	0,7772	51
39	0,7772	0,7753	0,7735	0,7716	0,7698	0,7679	0,7660	50
40	0,7660	0,7642	0,7623	0,7604	0,7585	0,7566	0,7547	49
41	0,7547	0,7528	0,7509	0,7490	0,7470	0,7451	0,7431	48
42	0,7431	0,7412	0,7392	0,7373	0,7353	0,7333	0,7314	47
43	0,7314	0,7294	0,7274	0,7254	0,7234	0,7214	0,7193	46
44	0,7193	0,7173	0,7153	0,7133	0,7112	0,7092	0,7071	45
45	0,7071	0,7051	0,7030	0,7009	0,6988	0,6968	0,6947	44
Degree	60′	50′	40′	30′	20′	10′	0′	Degree
	Sines							

N. B.—Decimal points indicated by commas.

Table No. III.—Sines and Cosines.

Degree	Sines							Degree
	0′	10′	20′	30′	40′	50′	60′	
0	0,0000	0,0029	0,0058	0,0087	0,0116	0,0145	0,0175	89
1	0,0175	0,0204	0,0233	0,0262	0,0291	0,0320	0,0349	88
2	0,0349	0,0378	0,0407	0,0436	0,0465	0,0494	0,0523	87
3	0,0523	0,0552	0,0581	0,0611	0,0640	0,0669	0,0698	86
4	0,0698	0,0727	0,0756	0,0785	0,0814	0,0843	0,0872	85
5	0,0872	0,0901	0,0930	0,0959	0,0987	0,1016	0,1045	84
6	0,1045	0,1074	0,1103	0,1132	0,1161	0,1190	0,1219	83
7	0,1219	0,1248	0,1276	0,1305	0,1334	0,1363	0,1392	82
8	0,1392	0,1421	0,1449	0,1478	0,1507	0,1536	0,1564	81
9	0,1564	0,1593	0,1622	0,1651	0,1679	0,1708	0,1737	80
10	0,1737	0,1765	0,1794	0,1822	0,1851	0,1880	0,1908	79
11	0,1908	0,1937	0,1965	0,1994	0,2022	0,2051	0,2079	78
12	0,2079	0,2108	0,2136	0,2164	0,2193	0,2221	0,2250	77
13	0,2250	0,2278	0,2306	0,2335	0,2363	0,2391	0,2419	76
14	0,2419	0,2447	0,2476	0,2504	0,2532	0,2560	0,2588	75
15	0,2588	0,2616	0,2644	0,2672	0,2700	0,2728	0,2756	74
16	0,2756	0,2784	0,2812	0,2840	0,2868	0,2896	0,2924	73
17	0,2924	0,2952	0,2979	0,3007	0,3035	0,3063	0,3090	72
18	0,3090	0,3118	0,3145	0,3173	0,3201	0,3228	0,3256	71
19	0,3256	0,3283	0,3311	0,3338	0,3366	0,3393	0,3420	70
20	0,3420	0,3448	0,3475	0,3502	0,3529	0,3557	0,3584	69
21	0,3584	0,3611	0,3638	0,3665	0,3692	0,3719	0,3746	68
22	0,3746	0,3773	0,3800	0,3827	0,3854	0,3881	0,3907	67
23	0,3907	0,3934	0,3961	0,3988	0,4014	0,4041	0,4067	66
24	0,4067	0,4094	0,4120	0,4147	0,4173	0,4200	0,4226	65
25	0,4226	0,4253	0,4279	0,4305	0,4331	0,4358	0,4384	64
26	0,4384	0,4410	0,4436	0,4462	0,4488	0,4514	0,4540	63
27	0,4540	0,4566	0,4592	0,4618	0,4643	0,4669	0,4695	62
28	0,4695	0,4720	0,4746	0,4772	0,4797	0,4823	0,4848	61
29	0,4848	0,4874	0,4899	0,4924	0,4950	0,4975	0,5000	60
30	0,5000	0,5025	0,5050	0,5075	0,5100	0,5125	0,5150	59
31	0,5150	0,5175	0,5200	0,5225	0,5250	0,5275	0,5299	58
32	0,5299	0,5324	0,5348	0,5373	0,5398	0,5422	0,5446	57
33	0,5446	0,5471	0,5495	0,5519	0,5544	0,5568	0,5592	56
34	0,5592	0,5616	0,5640	0,5664	0,5688	0,5712	0,5736	55
35	0,5736	0,5760	0,5783	0,5807	0,5831	0,5854	0,5878	54
36	0,5878	0,5901	0,5925	0,5948	0,5972	0,5995	0,6018	53
37	0,6018	0,6041	0,6065	0,6088	0,6111	0,6134	0,6157	52
38	0,6157	0,6180	0,6202	0,6225	0,6248	0,6271	0,6293	51
39	0,6293	0,6316	0,6338	0,6361	0,6383	0,6406	0,6428	50
40	0,6428	0,6450	0,6472	0,6495	0,6517	0,6539	0,6561	49
41	0,6561	0,6583	0,6604	0,6626	0,6648	0,6670	0,6691	48
42	0,6691	0,6713	0,6734	0,6756	0,6777	0,6799	0,6820	47
43	0,6820	0,6841	0,6862	0,6884	0,6905	0,6926	0,6947	46
44	0,6947	0,6968	0,6988	0,7009	0,7030	0,7051	0,7071	45
45	0,7071	0,7092	0,7112	0,7133	0,7153	0,7173	0,7193	44
Degree	60′	50′	40′	30′	20′	10′	0′	Degree
	Cosines							

N.B.—Decimal points indicated by commas.

Table No. IV.—Tangents and Cotangents.

Degree	Cotangents 0′	10′	20′	30′	40′	50′	60′	Degree
0	∞	343,774	171,885	114,589	85,9398	68,7501	57,2900	89
1	57,2900	49,1039	42,9641	38,1885	34,3678	31,2416	28,6362	88
2	28,6362	26,4316	24,5418	22,9038	21,4704	20,2056	19,0811	87
3	19,0811	18,0750	17,1693	16,3499	15,6048	14,9244	14,3007	86
4	14,3007	13,7267	13,1969	12,7062	12,2505	11,8262	11,4300	85
5	11,4300	11,0594	10,7119	10,3854	10,0780	9,7882	9,5144	84
6	9,5144	9,2553	9,0098	8,7769	8,5556	8,3450	8,1444	83
7	8,1444	7,9530	7,7704	7,5958	7,4287	7,2687	7,1154	82
8	7,1154	6,9682	6,8269	6,6912	6,5606	6,4348	6,3138	81
9	6,3138	6,1970	6,0844	5,9758	5,8708	5,7694	5,6713	80
10	5,6713	5,5764	5,4845	5,3955	5,3093	5,2257	5,1446	79
11	5,1446	5,0658	4,9894	4,9152	4,8430	4,7729	4,7046	78
12	4,7046	4,6383	4,5736	4,5107	4,4494	4,3897	4,3315	77
13	4,3315	4,2747	4,2193	4,1653	4,1126	4,0611	4,0108	76
14	4,0108	3,9617	3,9136	3,8667	3,8208	3,7760	3,7321	75
15	3,7321	3,6891	3,6471	3,6059	3,5656	3,5261	3,4874	74
16	3,4874	3,4495	3,4124	3,3759	3,3402	3,3052	3,2709	73
17	3,2709	3,2371	3,2041	3,1716	3,1397	3,1084	3,0777	72
18	3,0777	3,0475	3,0178	2,9887	2,9600	2,9319	2,9042	71
19	2,9042	2,8770	2,8502	2,8239	2,7980	2,7725	2,7475	70
20	2,7475	2,7228	2,6985	2,6746	2,6511	2,6279	2,6051	69
21	2,6051	2,5826	2,5605	2,5387	2,5172	2,4960	2,4751	68
22	2,4751	2,4545	2,4342	2,4142	2,3945	2,3750	2,3559	67
23	2,3559	2,3369	2,3183	2,2998	2,2817	2,2637	2,2460	66
24	2,2460	2,2286	2,2113	2,1943	2,1775	2,1609	2,1445	65
25	2,1445	2,1283	2,1123	2,0965	2,0809	2,0655	2,0503	64
26	2,0503	2,0353	2,0204	2,0057	1,9912	1,9768	1,9626	63
27	1,9626	1,9486	1,9347	1,9210	1,9074	1,8940	1,8807	62
28	1,8807	1,8676	1,8546	1,8418	1,8291	1,8165	1,8041	61
29	1,8041	1,7917	1,7796	1,7675	1,7556	1,7438	1,7321	60
30	1,7321	1,7205	1,7090	1,6977	1,6864	1,6753	1,6643	59
31	1,6643	1,6534	1,6426	1,6319	1,6213	1,6107	1,6003	58
32	1,6003	1,5900	1,5798	1,5697	1,5597	1,5497	1,5399	57
33	1,5399	1,5301	1,5204	1,5108	1,5013	1,4919	1,4826	56
34	1,4826	1,4733	1,4641	1,4550	1,4460	1,4370	1,4282	55
35	1,4282	1,4193	1,4106	1,4020	1,3934	1,3848	1,3764	54
36	1,3764	1,3680	1,3597	1,3514	1,3432	1,3351	1,3270	53
37	1,3270	1,3190	1,3111	1,3032	1,2954	1,2876	1,2799	52
38	1,2799	1,2723	1,2647	1,2572	1,2497	1,2423	1,2349	51
39	1,2349	1,2276	1,2203	1,2131	1,2059	1,1988	1,1918	50
40	1,1918	1,1847	1,1778	1,1709	1,1640	1,1572	1,1504	49
41	1,1504	1,1436	1,1369	1,1303	1,1237	1,1171	1,1106	48
42	1,1106	1,1041	1,0977	1,0913	1,0850	1,0786	1,0724	47
43	1,0724	1,0661	1,0599	1,0538	1,0477	1,0416	1,0355	46
44	1,0355	1,0295	1,0236	1,0176	1,0117	1,0058	1,0000	45
45	1,0000	0,9942	0,9884	0,9827	0,9770	0,9713	0,9657	44
Degree	60′	50′	40′	30′	20′	10′	0′	Degree
	Tangents							

N.B.—Decimal points indicated by commas.

Table No. IV.—Tangents and Cotangents.

Degree	Tangents							Degree
	0′	10′	20′	30′	40′	50′	60′	
0	0,0000	0,0029	0,0058	0,0087	0,0116	0,0145	0,0175	89
1	0,0175	0,0204	0,0233	0,0262	0,0291	0,0320	0,0349	88
2	0,0349	0,0378	0,0408	0,0437	0,0466	0,0495	0,0524	87
3	0,0524	0,0553	0,0582	0,0612	0,0641	0,0670	0,0699	86
4	0,0699	0,0729	0,0758	0,0787	0,0816	0,0846	0,0875	85
5	0,0875	0,0904	0,0934	0,0963	0,0992	0,1022	0,1051	84
6	0,1051	0,1081	0,1110	0,1139	0,1169	0,1198	0,1228	83
7	0,1228	0,1257	0,1287	0,1317	0,1346	0,1376	0,1405	82
8	0,1405	0,1435	0,1465	0,1495	0,1524	0,1554	0,1584	81
9	0,1584	0,1614	0,1644	0,1673	0,1703	0,1733	0,1763	80
10	0,1763	0,1793	0,1823	0,1853	0,1884	0,1914	0,1944	79
11	0,1944	0,1974	0,2004	0,2035	0,2065	0,2095	0,2126	78
12	0,2126	0,2156	0,2186	0,2217	0,2248	0,2278	0,2309	77
13	0,2309	0,2339	0,2370	0,2401	0,2432	0,2462	0,2493	76
14	0,2493	0,2524	0,2555	0,2586	0,2617	0,2648	0,2680	75
15	0,2680	0,2711	0,2742	0,2773	0,2805	0,2836	0,2868	74
16	0,2868	0,2899	0,2931	0,2962	0,2994	0,3026	0,3057	73
17	0,3057	0,3089	0,3121	0,3153	0,3185	0,3217	0,3249	72
18	0,3249	0,3281	0,3314	0,3346	0,3378	0,3411	0,3443	71
19	0,3443	0,3476	0,3509	0,3541	0,3574	0,3607	0,3640	70
20	0,3640	0,3673	0,3706	0,3739	0,3772	0,3805	0,3839	69
21	0,3839	0,3872	0,3906	0,3939	0,3973	0,4007	0,4040	68
22	0,4040	0,4074	0,4108	0,4142	0,4176	0,4211	0,4245	67
23	0,4245	0,4279	0,4314	0,4348	0,4383	0,4418	0,4452	66
24	0,4452	0,4487	0,4522	0,4557	0,4592	0,4628	0,4663	65
25	0,4663	0,4699	0,4734	0,4770	0,4806	0,4841	0,4877	64
26	0,4877	0,4913	0,4950	0,4986	0,5022	0,5059	0,5095	63
27	0,5095	0,5132	0,5169	0,5206	0,5243	0,5280	0,5317	62
28	0,5317	0,5355	0,5392	0,5430	0,5467	0,5505	0,5543	61
29	0,5543	0,5581	0,5619	0,5658	0,5696	0,5735	0,5774	60
30	0,5774	0,5812	0,5851	0,5891	0,5930	0,5969	0,6009	59
31	0,6009	0,6048	0,6088	0,6128	0,6168	0,6208	0,6249	58
32	0,6249	0,6289	0,6330	0,6371	0,6412	0,6453	0,6494	57
33	0,6494	0,6536	0,6577	0,6619	0,6661	0,6703	0,6745	56
34	0,6745	0,6788	0,6830	0,6873	0,6916	0,6959	0,7002	55
35	0,7002	0,7046	0,7089	0,7133	0,7177	0,7221	0,7265	54
36	0,7265	0,7310	0,7355	0,7400	0,7445	0,7490	0,7536	53
37	0,7536	0,7581	0,7627	0,7673	0,7720	0,7766	0,7813	52
38	0,7813	0,7860	0,7907	0,7954	0,8002	0,8050	0,8098	51
39	0,8098	0,8146	0,8195	0,8243	0,8292	0,8342	0,8391	50
40	0,8391	0,8441	0,8491	0,8541	0,8591	0,8642	0,8693	49
41	0,8693	0,8744	0,8796	0,8847	0,8899	0,8952	0,9004	48
42	0,9004	0,9057	0,9110	0,9163	0,9217	0,9271	0,9325	47
43	0,9325	0,9380	0,9435	0,9490	0,9545	0,9601	0,9657	46
44	0,9657	0,9713	0,9770	0,9827	0,9884	0,9942	1,0000	45
45	1,0000	1,0058	1,0117	1,0176	1,0236	1 0295	1,0355	44
Degree	60′	50′	40′	30′	20′	10′	0′	Degree
	Cotangents							

N.B.—Decimal points indicated by commas.

Table No. V.—Various Equivalents.

$\pi = 3{,}1415926586 \backsim 3{,}1416$	$\frac{\pi}{4} = 0{,}785398$	$g = 32{,}2$	$\sqrt{2g} = 8{,}025$
$\log \pi = 0{,}4971498727 \backsim 0{,}49715$	$\sqrt{\pi} = 1{,}772454$	$\log g = 1{,}5078559$	$e = 2{,}718282$
$\pi^2 = 9{,}869604$		$g^2 = 1036{,}84$	$\log e = 0{,}434295$
			$\log_e x =$ common $\log x \times 2{,}3$

Table No. VI. $\cos \omega + \lambda \cos 2\omega$.

Angle of Crank ω	$\cos \omega$	$\cos 2\omega$	$\lambda = \frac{R}{L} = \frac{1}{4}$	$\lambda = \frac{R}{L} = \frac{1}{4{,}5}$	$\lambda = \frac{R}{L} = \frac{1}{5}$	$\lambda = \frac{1}{4}$	$\lambda = \frac{1}{4{,}5}$	$\lambda = \frac{1}{5}$	Angle of Crank ω
			$\lambda \cdot \cos 2\omega$	$\lambda \cdot \cos 2\omega$	$\lambda \cdot \cos 2\omega$	$\cos \omega + \lambda \cos 2\omega$	$\cos \omega + \lambda \cos 2\omega$	$\cos \omega + \lambda \cos 2\omega$	
0°	+ 1,000	+ 1,000	+ 0,250	+ 0,222	+ 0,200	+ 1,250	+ 1,222	+ 1,200	0°
15°	+ 0,966	+ 0,866	+ 0,217	+ 0,192	+ 0,173	+ 1,183	+ 1,158	+ 1,139	15°
30°	+ 0,866	+ 0,500	+ 0,125	+ 0,111	+ 0,100	+ 0,991	+ 0,977	+ 0,966	30°
45°	+ 0,707	0,000	0,000	0,000	0,000	+ 0,707	+ 0,707	+ 0,707	45°
60°	+ 0,500	− 0,500	− 0,125	− 0,111	− 0,100	+ 0,375	+ 0,389	+ 0,40	60°
75°	+ 0,259	− 0,866	− 0,217	− 0,192	− 0,173	+ 0,042	+ 0,067	+ 0,086	75°
90°	0,000	− 1,000	− 0,250	− 0,222	− 0,200	− 0,250	− 0,222	− 0,200	90°
105°	− 0,259	− 0,866	− 0,217	− 0,192	− 0,173	− 0,476	− 0,451	− 0,432	105°
120°	− 0,500	− 0,500	− 0,125	− 0,111	− 0,100	− 0,625	− 0,611	− 0,600	120°
135°	− 0,707	0,000	0,000	0,000	0,000	− 0,707	− 0,707	− 0,707	135°
150°	− 0,866	+ 0,500	+ 0,125	+ 0,111	+ 0,100	− 0,741	− 0,755	− 0,766	150°
165°	− 0,966	+ 0,866	+ 0,217	+ 0,192	+ 0,173	− 0,749	− 0,774	− 0,793	165°
180°	− 1,00	+ 1,000	+ 0,250	+ 0,222	+ 0,200	− 0,750	− 0,778	− 0,800	180°

N.B.—Decimal points indicated by commas.

Table No. VII.

Inches and Millimetres.

Inches	mm	Inches	mm	Inches	mm	Inches	mm	Inches	mm
1/8	3,175	5 1/8	130,17	10 1/8	257,17	15 1/8	384,17	20 1/8	511,17
1/4	6,350	5 1/4	133,35	10 1/4	260,35	15 1/4	387,34	20 1/4	514,34
3/8	9,525	5 3/8	136,52	10 3/8	263,52	15 3/8	390,52	20 3/8	517,52
1/2	12,700	5 1/2	139,70	10 1/2	266,70	15 1/2	393,69	20 1/2	520,69
5/8	15,875	5 5/8	142,87	10 5/8	269,87	15 5/8	396,87	20 5/8	523,87
3/4	19,050	5 3/4	146,05	10 3/4	273,05	15 3/4	400,04	20 3/4	527,04
7/8	22,225	5 7/8	149,22	10 7/8	276,22	15 7/8	403,22	20 7/8	530,22
1	25,400	6	152,40	11	279,39	16	406,39	21	533,39
1 1/8	28,574	6 1/8	155,57	11 1/8	282,57	16 1/8	409,57	21 1/8	536,57
1 1/4	31,749	6 1/4	158,75	11 1/4	285,74	16 1/4	412,74	21 1/4	539,74
1 3/8	34,924	6 3/8	161,92	11 3/8	288,92	16 3/8	415,92	21 3/8	542,92
1 1/2	38,099	6 1/2	165,10	11 1/2	292,09	16 1/2	419,09	21 1/2	546,09
1 5/8	41,274	6 5/8	168,27	11 5/8	295,27	16 5/8	422,27	21 5/8	549,27
1 3/4	44,449	6 3/4	171,45	11 3/4	298,44	16 3/4	425,44	21 3/4	552,44
1 7/8	47,624	6 7/8	174,62	11 7/8	301,62	16 7/8	428,62	21 7/8	555,61
2	50,799	7	177,80	12	304,79	17	431,79	22	558,79
2 1/8	53,974	7 1/8	180,97	12 1/8	307,97	17 1/8	434,97	22 1/8	561,96
2 1/4	57,149	7 1/4	184,15	12 1/4	311,14	17 1/4	438,14	22 1/4	565,14
2 3/8	60,324	7 3/8	187,32	12 3/8	314,32	17 3/8	441,32	22 3/8	568,31
2 1/2	63,499	7 1/2	190,50	12 1/2	317,49	17 1/2	444,49	22 1/2	571,49
2 5/8	66,674	7 5/8	193,67	12 5/8	320,67	17 5/8	447,67	22 5/8	574,66
2 3/4	69,849	7 3/4	196,85	12 3/4	323,84	17 3/4	450,84	22 3/4	577,84
2 7/8	73,024	7 7/8	200,02	12 7/8	327,02	17 7/8	454,02	22 7/8	581,01
3	76,199	8	203,20	13	330,19	18	457,19	23	584,19
3 1/8	79,374	8 1/8	206,37	13 1/8	333,37	18 1/8	460,37	23 1/8	587,36
3 1/4	82,549	8 1/4	209,55	13 1/4	336,54	18 1/4	463,54	23 1/4	590,54
3 3/8	85,723	8 3/8	212,72	13 3/8	339,72	18 3/8	466,72	23 3/8	593,71
3 1/2	88,898	8 1/2	215,90	13 1/2	342,89	18 1/2	469,89	23 1/2	596,89
3 5/8	92,073	8 5/8	219,07	13 5/8	346,07	18 5/8	473,07	23 5/8	600,06
3 3/4	95,248	8 3/4	222,25	13 3/4	349,24	18 3/4	476,24	23 3/4	603,24
3 7/8	98,423	8 7/8	225,42	13 7/8	352,42	18 7/8	479,42	23 7/8	606,41
4	101,60	9	228,60	14	355,59	19	482,59	24	609,59
4 1/8	104,77	9 1/8	231,77	14 1/8	358,77	19 1/8	485,77	24 1/8	612,76
4 1/4	107,95	9 1/4	234,95	14 1/4	361,94	19 1/4	488,94	24 1/4	615,94
4 3/8	111,12	9 3/8	238,12	14 3/8	365,12	19 3/8	492,12	24 3/8	619,11
4 1/2	114,30	9 1/2	241,30	14 1/2	368,29	19 1/2	495,29	24 1/2	622,29
4 5/8	117,47	9 5/8	244,47	14 5/8	371,47	19 5/8	498,47	24 5/8	625,46
4 3/4	120,65	9 3/4	247,65	14 3/4	374,64	19 3/4	501,64	24 3/4	628,64
4 7/8	123,82	9 7/8	250,82	14 7/8	377,82	19 7/8	504,82	24 7/8	631,81
5	127,00	10	254,00	15	380,99	20	507,99	25	634,99

N.B.—Decimal points indicated by commas.

Table No. VII.

Inches and Millimetres.

Inches	mm	Inches	mm	Inches	mm	Inches	mm	Inches	mm
25 1/8	638,16	30 1/8	765,16	35 1/8	892,16	40 1/8	1019,2	45 1/8	1146,2
25 1/4	641,34	30 1/4	768,34	35 1/4	895,33	40 1/4	1022,3	45 1/4	1149,3
25 3/8	644,51	30 3/8	771,51	35 3/8	898,51	40 3/8	1025,5	45 3/8	1152,5
25 1/2	647,69	30 1/2	774,69	35 1/2	901,68	40 1/2	1028,7	45 1/2	1155,7
25 5/8	650,86	30 5/8	777,86	35 5/8	904,86	40 5/8	1031,9	45 5/8	1158,9
25 3/4	654,04	30 3/4	781,04	35 3/4	908,03	40 3/4	1035,0	45 3/4	1162,0
25 7/8	657,21	30 7/8	784,21	35 7/8	911,21	40 7/8	1038,2	45 7/8	1165,2
26	660,39	31	787,39	36	914,38	41	1041,4	46	1168,4
26 1/8	663,56	31 1/8	790,56	36 1/8	917,56	41 1/8	1044,6	46 1/8	1171,6
26 1/4	666,74	31 1/4	793,74	36 1/4	920,73	41 1/4	1047,7	46 1/4	1174,7
26 3/8	669,91	31 3/8	796,91	36 3/8	923,91	41 3/8	1050,9	46 3/8	1177,9
26 1/2	673,09	31 1/2	800,09	36 1/2	927,08	41 1/2	1054,1	46 1/2	1181,1
26 5/8	676,26	31 5/8	803,26	36 5/8	930,26	41 5/8	1057,3	46 5/8	1184,3
26 3/4	679,44	31 3/4	806,44	36 3/4	933,43	41 3/4	1060,4	46 3/4	1187,4
26 7/8	682,61	31 7/8	809,61	36 7/8	936,61	41 7/8	1063,6	46 7/8	1190,6
27	685,79	32	812,79	37	939,78	42	1066,8	47	1193,8
27 1/8	688,96	32 1/8	815,96	37 1/8	942,96	42 1/8	1070,0	47 1/8	1197,0
27 1/4	692,14	32 1/4	819,14	37 1/4	946,13	42 1/4	1073,1	47 1/4	1200,1
27 3/8	695,31	32 3/8	822,31	37 3/8	949,31	42 3/8	1076,3	47 3/8	1203,3
27 1/2	698,49	32 1/2	825,49	37 1/2	952,48	42 1/2	1079,5	47 1/2	1206,5
27 5/8	701,66	32 5/8	828,66	37 5/8	955,66	42 5/8	1082,7	47 5/8	1209,7
27 3/4	704,84	32 3/4	831,83	37 3/4	958,83	42 3/4	1085,8	47 3/4	1212,8
27 7/8	708,01	32 7/8	835,01	37 7/8	962,01	42 7/8	1089,0	47 7/8	1216,0
28	711,19	33	838,18	38	965,18	43	1092,2	48	1219,2
28 1/8	714,36	33 1/8	841,36	38 1/8	968,36	43 1/8	1095,4	48 1/8	1222,4
28 1/4	717,54	33 1/4	844,53	38 1/4	971,53	43 1/4	1098,5	48 1/4	1225,5
28 3/8	720,71	33 3/8	847,71	38 3/8	974,71	43 3/8	1101,7	48 3/8	1228,7
28 1/2	723,89	33 1/2	850,88	38 1/2	977,88	43 1/2	1104,9	48 1/2	1231,9
28 5/8	727,06	33 5/8	854,06	38 5/8	981,06	43 5/8	1108,1	48 5/8	1235,1
28 3/4	730,24	33 3/4	857,23	38 3/4	984,23	43 3/4	1111,2	48 3/4	1238,2
28 7/8	733,41	33 7/8	860,41	38 7/8	987,41	43 7/8	1114,4	48 7/8	1241,4
29	736,59	34	863,58	39	990,58	44	1117,6	49	1244,6
29 1/8	739,76	34 1/8	866,76	39 1/8	993,76	44 1/8	1120,8	49 1/8	1247,8
29 1/4	742,94	34 1/4	869,93	39 1/4	996,93	44 1/4	1123,9	49 1/4	1250,9
29 3/8	746,11	34 3/8	873,11	39 3/8	1000,1	44 3/8	1127,1	49 3/8	1254,1
29 1/2	749,29	34 1/2	876,28	39 1/2	1003,3	44 1/2	1130,3	49 1/2	1257,3
29 5/8	752,46	34 5/8	879,46	39 5/8	1006,5	44 5/8	1133,5	49 5/8	1260,5
29 3/4	755,64	34 3/4	882,63	39 3/4	1009,6	44 3/4	1136,6	49 3/4	1263,6
29 7/8	758,81	34 7/8	885,81	39 7/8	1012,8	44 7/8	1139,8	49 7/8	1266,8
30	761,99	35	888,98	40	1016,0	45	1143,0	50	1270,0

N.B.—Decimal points indicated by commas.

Table No. VII.

Inches and Millimetres.

Inches	mm	Inches	mm	Inches	mm	Inches	mm	Inches	mm
50 1/8	1273,2	55 1/8	1400,1	60 1/8	1527,1	65 1/8	1654,1	70 1/8	1781,1
50 1/4	1276,3	55 1/4	1403,3	60 1/4	1530,3	65 1/4	1657,3	70 1/4	1784,3
50 3/8	1279,5	55 3/8	1406,5	60 3/8	1533,5	65 3/8	1660,5	70 3/8	1787,5
50 1/2	1282,7	55 1/2	1409,7	60 1/2	1536,7	65 1/2	1663,7	70 1/2	1790,7
50 5/8	1285,9	55 5/8	1412,8	60 5/8	1539,8	65 5/8	1666,8	70 5/8	1793,8
50 3/4	1289,0	55 3/4	1416,0	60 3/4	1543,0	65 3/4	1670,0	70 3/4	1797,0
50 7/8	1292,2	55 7/8	1419,2	60 7/8	1546,2	65 7/8	1673,2	70 7/8	1800,2
51	1295,4	56	1422,4	61	1549,4	66	1676,4	71	1803,4
51 1/8	1298,6	56 1/8	1425,5	61 1/8	1552,5	66 1/8	1679,5	71 1/8	1806,5
51 1/4	1301,7	56 1/4	1428,7	61 1/4	1555,7	66 1/4	1682,7	71 1/4	1809,7
51 3/8	1304,9	56 3/8	1432,0	61 3/8	1558,9	66 3/8	1685,9	71 3/8	1812,9
51 1/2	1308,1	56 1/2	1435,1	61 1/2	1562,1	66 1/2	1689,1	71 1/2	1816,1
51 5/8	1311,3	56 5/8	1438,2	61 5/8	1565,2	66 5/8	1692,2	71 5/8	1819,2
51 3/4	1314,4	56 3/4	1441,4	61 3/4	1568,4	66 3/4	1695,4	71 3/4	1822,4
51 7/8	1317,6	56 7/8	1444,6	61 7/8	1571,6	66 7/8	1698,6	71 7/8	1825,6
52	1320,8	57	1447,8	62	1574,8	67	1701,8	72	1828,8
52 1/8	1324,0	57 1/8	1450,9	62 1/8	1577,9	67 1/8	1704,9	72 1/8	1831,9
52 1/4	1327,1	57 1/4	1454,1	62 1/4	1581,1	67 1/4	1708,1	72 1/4	1835,1
52 3/8	1330,3	57 3/8	1457,3	62 3/8	1584,3	67 3/8	1711,3	72 3/8	1838,3
52 1/2	1333,5	57 1/2	1460,5	62 1/2	1587,5	67 1/2	1714,5	72 1/2	1841,5
52 5/8	1336,7	57 5/8	1463,6	62 5/8	1590,6	67 5/8	1717,6	72 5/8	1844,6
52 3/4	1339,8	57 3/4	1466,8	62 3/4	1593,8	67 3/4	1720,8	72 3/4	1847,8
52 7/8	1343,0	57 7/8	1470,0	62 7/8	1597,0	67 7/8	1724,0	72 7/8	1851,0
53	1346,2	58	1473,2	63	1600,2	68	1727,2	73	1854,2
53 1/8	1349,4	58 1/8	1476,3	63 1/8	1603,3	68 1/8	1730,3	73 1/8	1857,3
53 1/4	1352,5	58 1/4	1479,5	63 1/4	1606,5	68 1/4	1733,5	73 1/4	1860,5
53 3/8	1355,7	58 3/8	1482,7	63 3/8	1609,7	68 3/8	1736,7	73 3/8	1863,7
53 1/2	1358,9	58 1/2	1485,9	63 1/2	1612,9	68 1/2	1739,9	73 1/2	1866,9
53 5/8	1362,1	58 5/8	1489,0	63 5/8	1616,0	68 5/8	1743,0	73 5/8	1870,0
53 3/4	1365,2	58 3/4	1492,2	63 3/4	1619,2	68 3/4	1746,2	73 3/4	1873,2
53 7/8	1368,4	58 7/8	1495,4	63 7/8	1622,4	68 7/8	1749,4	73 7/8	1876,4
54	1371,6	59	1498,5	64	1625,6	69	1752,6	74	1879,6
54 1/8	1374,8	59 1/8	1501,7	64 1/8	1628,7	69 1/8	1755,7	74 1/8	1882,7
54 1/4	1377,9	59 1/4	1504,9	64 1/4	1631,9	69 1/4	1758,9	74 1/4	1885,9
54 3/8	1381,1	59 3/8	1508,1	64 3/8	1635,1	69 3/8	1762,1	74 3/8	1889,1
54 1/2	1384,3	59 1/2	1511,3	64 1/2	1638,3	69 1/2	1765,3	74 1/2	1892,3
54 5/8	1387,4	59 5/8	1514,4	64 5/8	1641,4	69 5/8	1768,4	74 5/8	1895,4
54 3/4	1390,6	59 3/4	1517,6	64 3/4	1644,6	69 3/4	1771,6	74 3/4	1898,6
54 7/8	1393,8	59 7/8	1520,8	64 7/8	1647,8	69 7/8	1774,8	74 7/8	1901,8
55	1397,0	60	1524,0	65	1651,0	70	1778,0	75	1905,0

N. B.—Decimal points indicated by commas.

Table No. VII.

Inches and Millimetres.

Inches	mm	Inches	mm	Inches	mm	Inches	mm	Inches	mm
75 1/8	1908,1	80 1/8	2035,2	85 1/8	2162,2	90 1/8	2289,2	95 1/8	2416,2
75 1/4	1911,3	80 1/4	2038,4	85 1/4	2165,4	90 1/4	2292,4	95 1/4	2419,4
75 3/8	1914,5	80 3/8	2041,4	85 3/8	2168,4	90 3/8	2295,4	95 3/8	2422,4
75 1/2	1917,7	80 1/2	2044,6	85 1/2	2171,6	90 1/2	2298,6	95 1/2	2425,6
75 5/8	1920,8	80 5/8	2047,8	85 5/8	2174,8	90 5/8	2301,8	95 5/8	2428,8
75 3/4	1924,0	80 3/4	2051,0	85 3/4	2178,0	90 3/4	2305,0	95 3/4	2432,0
75 7/8	1927,2	80 7/8	2054,2	85 7/8	2181,2	90 7/8	2308,2	95 7/8	2435,2
76	1930,4	81	2057,4	86	2184,4	91	2311,4	96	2438,4
76 1/8	1933,5	81 1/8	2060,6	86 1/8	2187,6	91 1/8	2314,6	96 1/8	2441,6
76 1/4	1936,7	81 1/4	2063,8	86 1/4	2190,8	91 1/4	2317,8	96 1/4	2444,8
76 3/8	1939,9	81 3/8	2066,8	86 3/8	2193,8	91 3/8	2320,8	96 3/8	2447,8
76 1/2	1943,1	81 1/2	2070,0	86 1/2	2197,0	91 1/2	2324,0	96 1/2	2451,0
76 5/8	1946,2	81 5/8	2073,2	86 5/8	2200,2	91 5/8	2327,2	96 5/8	2454,2
76 3/4	1949,4	81 3/4	2076,4	86 3/4	2203,4	91 3/4	2330,4	96 3/4	2457,4
76 7/8	1952,6	81 7/8	2079,6	86 7/8	2206,6	91 7/8	2333,6	96 7/8	2460,6
77	1955,8	82	2082,8	87	2209,8	92	2336,8	97	2463,8
77 1/8	1958,9	82 1/8	2086,0	87 1/8	2213,0	92 1/8	2340,0	97 1/8	2467,0
77 1/4	1962,1	82 1/4	2089,2	87 1/4	2216,2	92 1/4	2343,2	97 1/4	2470,2
77 3/8	1965,3	82 3/8	2092,2	87 3/8	2219,2	92 3/8	2346,2	97 3/8	2473,2
77 1/2	1968,5	82 1/2	2095,4	87 1/2	2222,4	92 1/2	2349,4	97 1/2	2476,4
77 5/8	1971,6	82 5/8	2098,6	87 5/8	2225,6	92 5/8	2352,6	97 5/8	2479,6
77 3/4	1974,8	82 3/4	2101,8	87 3/4	2228,8	92 3/4	2355,8	97 3/4	2482,8
77 7/8	1978,0	82 7/8	2105,0	87 7/8	2232,0	92 7/8	2359,0	97 7/8	2486,0
78	1981,2	83	2108,2	88	2235,2	93	2362,2	98	2489,2
78 1/8	1984,3	83 1/8	2111,4	88 1/8	2238,4	93 1/8	2365,4	98 1/8	2492,4
78 1/4	1987,5	83 1/4	2114,6	88 1/4	2241,6	93 1/4	2368,6	98 1/4	2495,6
78 3/8	1990,7	83 3/8	2117,6	88 3/8	2244,6	93 3/8	2371,6	98 3/8	2498,6
78 1/2	1993,9	83 1/2	2120,8	88 1/2	2247,8	93 1/2	2374,8	98 1/2	2501,8
78 5/8	1997,0	83 5/8	2124,0	88 5/8	2251,0	93 5/8	2378,0	98 5/8	2505,0
78 3/4	2000,2	83 3/4	2127,2	88 3/4	2254,2	93 3/4	2381,2	98 3/4	2508,2
78 7/8	2003,4	83 7/8	2130,4	88 7/8	2257,4	93 7/8	2384,4	98 7/8	2511,4
79	2006,6	84	2133,6	89	2260,6	94	2387,6	99	2514,6
79 1/8	2009,8	84 1/8	2136,8	89 1/8	2263,8	94 1/8	2390,8	99 1/8	2517,8
79 1/4	2013,0	84 1/4	2140,0	89 1/4	2267,0	94 1/4	2394,0	99 1/4	2521,0
79 3/8	2016,0	84 3/8	2143,0	89 3/8	2270,0	94 3/8	2397,0	99 3/8	2524,0
79 1/2	2019,2	84 1/2	2146,2	89 1/2	2273,2	94 1/2	2400,2	99 1/2	2527,2
79 5/8	2022,4	84 5/8	2149,4	89 5/8	2276,4	94 5/8	2403,4	99 5/8	2530,4
79 3/4	2025,6	84 3/4	2152,6	89 3/4	2279,6	94 3/4	2406,6	99 3/4	2533,6
79 7/8	2028,8	84 7/8	2155,8	89 7/8	2282,8	94 7/8	2409,8	99 7/8	2536,8
80	2032,0	85	2159,0	90	2286,0	95	2413,0	100	2540,0

N.B.—Decimal points indicated by commas.

Table No. VIII.
Square Metres and Square Feet.

Square Metres	Square Feet	Square Metres	Square Feet	Square Metres	Square Feet	Square Metres	Square Feet
1	10,764	26	279,872	51	548,979	76	818,087
2	21,529	27	290,636	52	559,744	77	828,851
3	32,293	28	301,400	53	570,508	78	839,615
4	43,057	29	312,165	54	581,272	79	850,380
5	53,822	30	322,929	55	592,036	80	861,144
6	64,586	31	333,693	56	602,801	81	871,908
7	75,350	32	344,458	57	613,565	82	882,673
8	86,114	33	355,222	58	624,329	83	893,437
9	96,879	34	365,986	59	635,094	84	904,201
10	107,643	35	376,750	60	645,858	85	914,966
11	118,407	36	387,515	61	656,622	86	925,730
12	129,172	37	398,279	62	667,387	87	936,494
13	139,936	38	409,043	63	678,151	88	947,258
14	150,700	39	419,808	64	688,915	89	958,023
15	161,464	40	430,572	65	699,680	90	968,787
16	172,229	41	441,336	66	710,444	91	979,551
17	182,993	42	452,101	67	721,208	92	990,316
18	193,757	43	462,865	68	731,972	93	1001,080
19	204,522	44	473,629	69	742,737	94	1011,844
20	215,286	45	484,394	70	753,501	95	1022,608
21	226,050	46	495,158	71	764,265	96	1033,373
22	236,815	47	505,922	72	775,030	97	1044,137
23	247,579	48	516,686	73	785,794	98	1054,901
24	258,343	49	527,451	74	796,558	99	1065,666
25	269,108	50	538,215	75	807,322	100	1076,430

N.B.—Decimal points indicated by commas.

Table No. IX.
Square Feet and Square Metres.

Square Feet	Square Metres	Square Feet	Square Metres	Square Feet	Square Metres	Square Feet	Square Metres
1	0,0929	26	2,4154	51	4,7379	76	7,0604
2	0,1858	27	2,5083	52	4,8308	77	7,1533
3	0,2787	28	2,6012	53	4,9237	78	7,2462
4	0,3716	29	2,6941	54	5,0166	79	7,3391
5	0,4645	30	2,7870	55	5,1095	80	7,4320
6	0,5574	31	2,8799	56	5,2024	81	7,5249
7	0,6503	32	2,9728	57	5,2953	82	7,6178
8	0,7432	33	3,0657	58	5,3882	83	7,7107
9	0,8361	34	3,1586	59	5,4811	84	7,8036
10	0,9290	35	3,2515	60	5,5740	85	7,8965
11	1,0219	36	3,3444	61	5,6669	86	7,9894
12	1,1148	37	3,4373	62	5,7598	87	8,0823
13	1,2077	38	3,5302	63	5,8527	88	8,1752
14	1,3006	39	3,6231	64	5,9456	89	8,2681
15	1,3935	40	3,7160	65	6,0385	90	8,3610
16	1,4864	41	3,8089	66	6,1314	91	8,4539
17	1,5793	42	3,9018	67	6,2243	92	8,5468
18	1,6722	43	3,9947	68	6,3172	93	8,6397
19	1,7651	44	4,0876	69	6,4101	94	8,7326
20	1,8580	45	4,1805	70	6,5030	95	8,8255
21	1,9509	46	4,2734	71	6,5959	96	8,9184
22	2,0438	47	4,3663	72	6,6888	97	9,0113
23	2,1367	48	4,4592	73	6,7817	98	9,1042
24	2,2296	49	4,5521	74	6,8746	99	9,1971
25	2,3225	50	4,6450	75	6,9675	100	9,2900

N.B.—Decimal points indicated by commas.

Table No. XI.
Pounds and Kilogrammes.

Pounds	Kilogr.	Pounds	Kilogr.	Pounds	Kilogr.	Pounds	Kilogr.
1	0,4536	41	18,5973	81	36,7410	121	54,8847
2	0,9072	42	19,0509	82	37,1946	122	55,3383
3	1,3608	43	19,5045	83	37,6482	123	55,7919
4	1,8144	44	19,9581	84	38,1018	124	56,2455
5	2,2680	45	20,4117	85	38,5554	125	56,6991
6	2,7216	46	20,8653	86	39,0089	126	57,1527
7	3,1752	47	21,3189	87	39,4625	127	57,6063
8	3,6287	48	21,7724	88	39,9161	128	58,0599
9	4,0823	49	22,2260	89	40,3697	129	58,5135
10	4,5359	50	22,6796	90	40,8233	130	58,9671
11	4,9895	51	23,1332	91	41,2769	131	59,4207
12	5,4431	52	23,5868	92	41,7305	132	59,8742
13	5,8967	53	24,0404	93	42,1841	133	60,3278
14	6,3503	54	24,4940	94	42,6377	134	60,7814
15	6,8039	55	24,9476	95	43,0913	135	61,2350
16	7,2575	56	25,4012	96	43,5449	136	61,6885
17	7,7111	57	25,8548	97	43,9985	137	62,1421
18	8,1647	58	26,3084	98	44,4521	138	62,5958
19	8,6182	59	26,7619	99	44,9057	139	63,0494
20	9,0718	60	27,2155	100	45,3593	140	63,5030
21	9,5254	61	27,6691	101	45,8128	141	63,9566
22	9,9790	62	28,1227	102	46,2664	142	64,4102
23	10,4326	63	28,5763	103	46,7200	143	64,8638
24	10,8862	64	29,0299	104	47,1736	144	65,3174
25	11,3398	65	29,4835	105	47,6272	145	65,7710
26	11,7934	66	29,9371	106	48,0808	146	66,2246
27	12,2470	67	30,3907	107	48,5344	147	66,6782
28	12,7006	68	30,8443	108	48,9880	148	67,1317
29	13,1542	69	31,2979	109	49,4416	149	67,5853
30	13,6078	70	31,7515	110	49,8952	150	68,0389
31	14,0614	71	32,2051	111	50,3488	151	68,4925
32	14,5149	72	32,6587	112	50,8024	152	68,9461
33	14,9685	73	33,1123	113	51,2560	153	69,3997
34	15,4221	74	33,5658	114	51,7096	154	69,8533
35	15,8757	75	34,0194	115	52,1632	155	70,3069
36	16,3292	76	34,4730	116	52,5168	156	70,7605
37	16,7298	77	34,9266	117	53,0704	157	71,2141
38	17,2365	78	35,3802	118	53,5240	158	71,6677
39	17,6901	79	35,8338	119	53,9775	159	72,1212
40	18,1437	80	36,2874	120	54,4311	160	72,5748

N.B.—Decimal points indicated by commas.

Table No. XII.
Kilogrammes and Pounds.

Kilogr.	Pounds	Kilogr.	Pounds	Kilogr.	Pounds	Kilogr.	Pounds
1	2,2046	41	90,3895	81	178,5743	121	266,7591
2	4,4092	42	92,5941	82	180,7789	122	268,9638
3	6,6139	43	94,7987	83	182,9836	123	271,1684
4	8,8185	44	97,0034	84	185,1882	124	273,3730
5	11,0231	45	99,2079	85	187,3928	125	275,5776
6	13,2277	46	101,4126	86	189,5974	126	277,7823
7	15,4324	47	103,6172	87	191,8020	127	279,9869
8	17,6370	48	105,8218	88	194,0067	128	282,1915
9	19,8416	49	108,0264	89	196,2113	129	284,3961
10	22,0462	50	110,2311	90	198,4159	130	286,6004
11	24,2508	51	112,4357	91	200,6205	131	288,8054
12	26,4554	52	114,6403	92	202,8251	132	291,0100
13	28,6601	53	116,8499	93	205,0298	133	293,2146
14	30,8647	54	119,0495	94	207,2344	134	295,4192
15	33,0693	55	121,2542	95	209,4390	135	297,6238
16	35,2739	56	123,4588	96	211,6431	136	299,8285
17	37,4786	57	125,6634	97	213,8482	137	302,0330
18	39,6832	58	127,8680	98	216,0529	138	304,2337
19	41,8878	59	130,6727	99	218,2575	139	306,4423
20	44,0924	60	132,2773	100	220,4621	140	308,6469
21	46,2970	61	134,4819	101	222,6667	141	310,8516
22	48,5017	62	136,6865	102	224,8713	142	313,0562
23	50,7063	63	138,8911	103	227,0760	143	315,2608
24	52,9109	64	141,0958	104	229,2806	144	317,4655
25	55,1155	65	143,3004	105	231,4852	145	319,6700
26	57,3202	66	145,5050	106	233,6898	146	321,8747
27	59,5248	67	147,7096	107	235,8945	147	324,0793
28	61,7294	68	149,9142	108	238,0991	148	326,2839
29	63,9340	69	152,1189	109	240,3037	149	328,4885
30	66,1386	70	154,3235	110	242,5083	150	330,6932
31	68,3433	71	156,5281	111	244,7129	151	332,8978
32	70,5479	72	158,7327	112	246,9175	152	335,1024
33	72,7525	73	160,9374	113	249,1222	153	337,3120
34	74,9571	74	163,1419	114	251,3268	154	339,5116
35	77,1617	75	165,3466	115	253,5314	155	341,7163
36	79,3664	76	167,5512	116	255,7360	156	343,9209
37	81,5709	77	169,7559	117	257,9407	157	346,1254
38	83,7756	78	171,9605	118	260,1453	158	348,3301
39	85,9802	79	174,1651	119	262,3499	159	351,1348
40	88,1848	80	176,3697	120	264,5545	160	352,7394

N.B.—Decimal points indicated by commas.

Table No. XIII.

Pounds per Square Inch and Kilogrammes per Square Centimetre.

Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.
1	0,0703	36	2,530	71	4,991	106	7,452	141	9,913
2	0,1406	37	2,601	72	5,061	107	7,522	142	9,983
3	0,2109	38	2,671	73	5,131	108	7,593	143	10,054
4	0,2812	39	2,741	74	5,202	109	7,663	144	10,124
5	0,3515	40	2,812	75	5,272	110	7,733	145	10,194
6	0,4218	41	2,882	76	5,342	111	7,804	146	10,264
7	0,4921	42	2,952	77	5,413	112	7,874	147	10,335
8	0,5624	43	3,022	78	5,483	113	7,944	148	10,405
9	0,6327	44	3,093	79	5,553	114	8,015	149	10,475
10	0,7030	45	3,163	80	5,624	115	8,085	150	10,546
11	0,7733	46	3,233	81	5,694	116	8,155	155	10,897
12	0,8436	47	3,304	82	5,764	117	8,226	160	11,249
13	0,9140	48	3,374	83	5,834	118	8,296	165	11,600
14	0,9843	49	3,444	84	5,905	119	8,366	170	11,952
15	1,0546	50	3,515	85	5,975	120	8,436	175	12,303
16	1,1248	51	3,585	86	6,045	121	8,507	180	12,655
17	1,1952	52	3,655	87	6,116	122	8,577	185	13,006
18	1,2655	53	3,725	88	6,186	123	8,647	190	13,358
19	1,3351	54	3,796	89	6,256	124	8,718	195	13,710
20	1,406	55	3,866	90	6,327	125	8,788	200	14,061
21	1,476	56	3,936	91	6,397	126	8,858	210	14,76
22	1,546	57	4,007	92	6,467	127	8,929	220	15,46
23	1,616	58	4,077	93	6,537	128	8,999	230	16,16
24	1,687	59	4,147	94	6,608	129	9,069	240	16,87
25	1,757	60	4,218	95	6,678	130	9,140	250	17,57
26	1,827	61	4,288	96	6,748	131	9,210	260	18,27
27	1,898	62	4,358	97	6,819	132	9,280	270	18,98
28	1,968	63	4,428	98	6,889	133	9,350	280	19,68
29	2,038	64	4,499	99	6,959	134	9,421	290	20,38
30	2,109	65	4,569	100	7,030	135	9,491	300	21,09
31	2,179	66	4,639	101	7,101	136	9,561	310	21,79
32	2,249	67	4,710	102	7,171	137	9,632	320	22,49
33	2,319	68	4,780	103	7,241	138	9,702	330	23,19
34	2,390	69	4,850	104	7,312	139	9,772	340	23,90
35	2,460	70	4,921	105	7,382	140	9,843	350	24,60

N.B.—Decimal points indicated by commas.

Table No. XIV.

Kilogrammes per Square Centimetre and Pounds per Square Inch.

Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch	Kilogr. per sq. cm.	Pounds per square inch
0,1	1,422	3,6	51,203	7,1	100,984	10,6	150,77	14,1	200,55
0,2	2,844	3,7	52,625	7,2	102,407	10,7	152,19	14,2	201,97
0,3	4,267	3,8	54,048	7,3	103,829	10,8	153,61	14,3	203,39
0,4	5,689	3,9	55,470	7,4	105,251	10,9	155,03	14,4	204,81
0,5	7,111	4,0	56,892	7,5	106,674	11,0	156,46	14,5	206,24
0,6	8,533	4,1	58,315	7,6	108,096	11,1	157,88	14,6	207,65
0,7	9,956	4,2	59,737	7,7	109,518	11,2	159,30	14,7	209,08
0,8	11,378	4,3	61,159	7,8	110,940	11,3	160,72	14,8	210,50
0,9	12,800	4,4	62,582	7,9	112,363	11,4	162,14	14,9	211,92
1,0	14,223	4,5	64,004	8,0	113,785	11,5	163,57	15,0	213,35
1,1	15,645	4,6	65,426	8,1	115,207	11,6	164,99	15,1	214,77
1,2	17,067	4,7	66,849	8,2	116,630	11,7	166,41	15,2	216,19
1,3	18,490	4,8	68,271	8,3	118,052	11,8	167,83	15,3	217,61
1,4	19,912	4,9	69,693	8,4	119,474	11,9	169,26	15,4	219,04
1,5	21,334	5,0	71,116	8,5	120,897	12,0	170,68	15,5	220,46
1,6	22,757	5,1	72,538	8,6	122,319	12,1	172,10	15,6	221,88
1,7	24,179	5,2	73,960	8,7	123,741	12,2	173,52	15,7	223,30
1,8	25,601	5,3	75,382	8,8	125,164	12,3	174,94	15,8	224,73
1,9	27,024	5,4	76,805	8,9	126,586	12,4	176,37	15,9	226,15
2,0	28,446	5,5	78,227	9,0	128,008	12,5	177,79	16,0	227,57
2,1	29,868	5,6	79,649	9,1	129,431	12,6	179,21	16,5	234,68
2,2	31,291	5,7	81,072	9,2	130,853	12,7	180,63	17,0	241,79
2,3	32,713	5,8	82,494	9,3	132,275	12,8	182,06	17,5	248,91
2,4	34,135	5,9	83,916	9,4	133,698	12,9	183,48	18,0	256,02
2,5	35,558	6,0	85,339	9,5	135,120	13,0	184,90	18,5	263,13
2,6	36,980	6,1	86,761	9,6	136,542	13,1	186,32	19,0	270,24
2,7	38,402	6,2	88,183	9,7	137,965	13,2	187,75	19,5	277,35
2,8	39,824	6,3	89,606	9,8	139,387	13,3	189,17	20,0	284,46
2,9	41,247	6,4	91,028	9,9	140,809	13,4	190,59	20,5	291,58
3,0	42,669	6,5	92,450	10,0	142,230	13,5	192,01	21,0	298,69
3,1	44,091	6,6	93,873	10,1	143,650	13,6	193,43	21,5	305,80
3,2	45,514	6,7	95,295	10,2	145,080	13,7	194,86	22,0	312,91
3,3	46,936	6,8	96,717	10,3	146,500	13,8	196,28	22,5	320,02
3,4	48,358	6,9	98,140	10,4	147,920	13,9	197,70	23,0	327,13
3,5	49,781	7,0	99,562	10,5	149,340	14,0	199,12	23,5	334,25

N.B.—Decimal points indicated by commas.

Table No. XXI.

Comparison of Thermometers.

Centigrade	Réaumur	Fahrenheit	Centigrade	Réaumur	Fahrenheit
— 20	— 16	— 4	+ 20	+ 16,0	+ 68,0
— 19	— 15,2	— 2,2	+ 21	+ 16,8	+ 69,8
— 18	— 14,4	— 0,4	+ 22	+ 17,6	+ 71,6
— 17	— 13,6	+ 1,4	+ 23	+ 18,4	+ 73,4
— 16	— 12,8	+ 3,2	+ 24	+ 19,2	+ 75,2
— 15	— 12,0	+ 5,0	+ 25	+ 20,0	+ 77,0
— 14	— 11,2	+ 6,8	+ 26	+ 20,8	+ 78,8
— 13	— 10,4	+ 8,6	+ 27	+ 21,6	+ 80,6
— 12	— 9,6	+ 10,4	+ 28	+ 22,4	+ 82,4
— 11	— 8,8	+ 12,2	+ 29	+ 23,2	+ 84,2
— 10	— 8,0	+ 14,0	+ 30	+ 24,0	+ 86,0
— 9	— 7,2	+ 15,8	+ 31	+ 24,8	+ 87,8
— 8	— 6,4	+ 17,6	+ 32	+ 25,6	+ 89,6
— 7	— 5,6	+ 19,4	+ 33	+ 26,4	+ 91,4
— 6	— 4,8	+ 21,2	+ 34	+ 27,2	+ 93,2
— 5	— 4,0	+ 23,0	+ 35	+ 28,0	+ 95,0
— 4	— 3,2	+ 24,8	+ 36	+ 28,8	+ 96,8
— 3	— 2,4	+ 26,6	+ 37	+ 29,6	+ 98,6
— 2	— 1,6	+ 28,4	+ 38	+ 30,4	+ 100,4
— 1	— 0,8	+ 30,2	+ 39	+ 31,2	+ 102,2
0	0	+ 32,0	+ 40	+ 32,0	+ 104,0
+ 1	+ 0,8	+ 33,8	+ 41	+ 32,8	+ 105,8
+ 2	+ 1,6	+ 35,6	+ 42	+ 33,6	+ 107,6
+ 3	+ 2,4	+ 37,4	+ 43	+ 34,4	+ 109,4
+ 4	+ 3,2	+ 39,2	+ 44	+ 35,2	+ 111,2
+ 5	+ 4,0	+ 41,0	+ 45	+ 36,0	+ 113,0
+ 6	+ 4,8	+ 42,8	+ 46	+ 36,8	+ 114,8
+ 7	+ 5,6	+ 44,6	+ 47	+ 37,6	+ 116,6
+ 8	+ 6,4	+ 46,4	+ 48	+ 38,4	+ 118,4
+ 9	+ 7,2	+ 48,2	+ 49	+ 39,2	+ 120,2
+ 10	+ 8,0	+ 50,0	+ 50	+ 40,0	+ 122,0
+ 11	+ 8,8	+ 51,8	+ 51	+ 40,8	+ 123,8
+ 12	+ 9,6	+ 53,6	+ 52	+ 41,6	+ 125,6
+ 13	+ 10,4	+ 55,4	+ 53	+ 42,4	+ 127,4
+ 14	+ 11,2	+ 57,2	+ 54	+ 43,2	+ 129,2
+ 15	+ 12,0	+ 59,0	+ 55	+ 44,0	+ 131,0
+ 16	+ 12,8	+ 60,8	+ 56	+ 44,8	+ 132,8
+ 17	+ 13,6	+ 62,6	+ 57	+ 45,6	+ 134,6
+ 18	+ 14,4	+ 64,4	+ 58	+ 46,4	+ 136,4
+ 19	+ 15,2	+ 66,2	+ 59	+ 47,2	+ 138,2

N.B.—Decimal points indicated by commas.

Table No. XXI.—*continued.*

Comparison of Thermometers.

Centigrade	Réaumur	Fahrenheit	Centigrade	Réaumur	Fahrenheit
+60	+48,0	+140,0	+100	+ 80,0	+212,0
+61	+48,8	+141,8	+101	+ 80,8	+213,8
+62	+49,6	+143,6	+102	+ 81,6	+215,6
+63	+50,4	+145,4	+103	+ 82,4	+217,4
+64	+51,2	+147,2	+104	+ 83,2	+219,2
+65	+52,0	+149,0	+105	+ 84,0	+221,0
+66	+52,8	+150,8	+106	+ 84,8	+222,8
+67	+53,6	+152,6	+107	+ 85,6	+224,6
+68	+54,4	+154,4	+108	+ 86,4	+226,4
+69	+55,2	+156,2	+109	+ 87,2	+228,2
+70	+56,0	+158,0	+110	+ 88,0	+230,0
+71	+56,8	+159,8	+111	+ 88,8	+231,8
+72	+57,6	+161,6	+112	+ 89,6	+233,6
+73	+58,4	+163,4	+113	+ 90,4	+235,4
+74	+59,2	+165,2	+114	+ 91,2	+237,2
+75	+60,0	+167,0	+115	+ 92,0	+239,0
+76	+60,8	+168,8	+116	+ 92,8	+240,8
+77	+61,6	+170,6	+117	+ 93,6	+242,6
+78	+62,4	+172,4	+118	+ 94,4	+244,4
+79	+63,2	+174,2	+119	+ 95,2	+246,2
+80	+64,0	+176,0	+120	+ 96,0	+248,0
+81	+64,8	+177,8	+121	+ 96,8	+249,8
+82	+65,6	+179,6	+122	+ 97,6	+251,6
+83	+66,4	+181,4	+123	+ 98,4	+253,4
+84	+67,2	+183,2	+124	+ 99,2	+255,2
+85	+68,0	+185,0	+125	+100,0	+257,0
+86	+68,8	+186,8	+126	+100,8	+258,8
+87	+69,6	+188,6	+127	+101,6	+260,6
+88	+70,4	+190,4	+128	+102,4	+262,4
+89	+71,2	+192,2	+129	+103,2	+264,2
+90	+72,0	+194,0	+130	+104,0	+266,0
+91	+72,8	+195,8	+131	+104,8	+267,8
+92	+73,6	+197,6	+132	+105,6	+269,6
+93	+74,4	+199,4	+133	+106,4	+271,4
+94	+75,2	+201,2	+134	+107,2	+273,2
+95	+76,0	+203,0	+135	+108,0	+275,0
+96	+76,8	+204,8			
+97	+77,6	+206,6			
+98	+78,4	+208,4			
+99	+79,2	+210,2			

N.B.—Decimal points indicated by commas.

TABLE No. XXII.—*Properties of Saturated Steam.*

In this table Joule's equivalent of heat = 772·8 foot pounds per degree Fahrenheit.

1	2		3	4	5	6	7	8	9
Total Pressure. Pounds per square inch.	Temperature or Boiling Point in Degrees Fahr.		Total Heat required to raise 1 lb. of Water from 32° Fahr. to Boiling Point.	Heat absorbed per pound of Steam in overcoming *Internal* Resistance to Vaporisation.	Heat absorbed per pound of Steam in overcoming *External* Resistance to Expansion.	Total Heat required to generate 1 lb. of Steam from Water at 32° under Constant Pressure. (3+4+5).	Volume of Steam compared with Water at 32°; or cubic feet of Steam generated by 1 cubic foot of Water at 32°.	Volume. Cubic feet per pound.	Weight. Pounds per cubic foot.
Abs.	Fahr.	Abs.	Br. Th. U.	Br. Th. U.	Br. Th. U.	Br. Th. U.			
0·0	−461·2	0	—	—	—	—	—	—	—
1·00	102·0	563·2	70·03	980·97	62·05	1113·05	20,760	332·6	0·003007
1·5	115·8	577·0	83·85	969·94	63·47	1117·26	14,157	226·8	0·004410
2·0	126·4	587·6	94·47	961·49	64·53	1120·49	10,798	173·0	0·005781
2·5	134·8	596·0	102·89	954·81	65·36	1123·06	8,744	140·1	0·007144
3·0	141·6	602·8	109·71	949·35	66·08	1125·14	7,362	118·0	0·008476
3·5	147·8	609·0	115·93	944·40	66·69	1127·02	6,377	102·16	0·009788
4·0	153·1	614·3	121·25	940·14	67·24	1128·63	5,625	90·12	0·01109
4·5	157·8	619·0	125·97	936·38	67·72	1130·07	5,035	80·67	0·01240
5·0	162·3	623·5	130·49	932·82	68·13	1131·44	4,559	73·05	0·01369
5·5	166·4	627·6	134·61	929·58	68·50	1132·69	4,167	66·77	0·01498
6·0	170·1	631·3	138·33	926·66	68·83	1133·82	3,838	61·50	0·01626
6·5	173·5	634·7	141·75	923·97	69·14	1134·86	3,559	57·03	0·01754
7·0	176·9	638·1	145·17	921·32	69·41	1135·90	3,318	53·16	0·01881
7·5	180·0	641·2	148·29	918·89	69·66	1136·84	3,107	49·79	0·02009
8·0	183·0	644·2	151·31	916·56	69·88	1137·75	2,923	46·83	0·02135
9·0	188·4	649·6	156·74	912·37	70·29	1139·40	2,613	41·87	0·02389

10·0	193·3	654·5	161·67	908·57	70·65	1140·89	2,363	37·87	0·02641
11·0	197·8	659·0	166·20	905·08	70·98	1142·26	2,159	34·60	0·02887
12·0	202·0	663·2	170·43	901·83	71·29	1143·55	1,988	31·85	0·03140
13·0	205·9	667·1	174·36	898·80	71·56	1144·72	1,842	29·51	0·03389
14·0	209·6	670·8	178·09	895·95	71·83	1145·87	1,716	27·50	0·03636
14·7	212·0	673·2	180·52	893·81	72·27	1146·60	1,644	26·36	0·03794
15·0	213·1	674·3	181·62	892·96	72·35	1146·93	1,614	25·86	0·03867
16·0	216·3	677·5	184·85	890·46	72·60	1147·91	1,516	24·32	0·04112
17·0	219·5	680·7	188·08	887·97	72·84	1148·89	1,435	22·97	0·04354
18·0	222·5	683·7	191·12	885·60	73·08	1149·80	1,360	21·76	0·04595
19·0	225·3	686·5	193·86	883·46	73·30	1150·62	1,292	20·68	0·04835
20·0	228·0	689·2	196·68	881·27	73·52	1151·47	1,231	19·70	0·05676
21·0	230·7	691·9	199·41	879·16	73·73	1152·30	1,176	18·82	0·05314
22·0	233·3	694·5	202·04	877·12	73·93	1153·09	1,125	18·01	0·05553
23·0	235·8	697·0	204·57	875·16	74·12	1153·85	1,079	17·27	0·05791
24·0	238·2	699·4	207·00	873·26	74·32	1154·58	1,036	16·60	0·06024
25·0	240·5	701·7	209·33	871·46	74·50	1155·29	998	15·97	0·06262
26·0	242·7	703·9	211·56	869·72	74·68	1155·96	961	15·39	0·06498
27·0	244·8	706·0	213·69	868·06	74·85	1156·60	928	14·86	0·06730
28·0	246·8	708·0	215·71	866·48	75·02	1157·21	897	14·36	0·06962
29·0	248·7	709·9	217·64	864·96	75·19	1157·79	868	13·89	0·07200
30·0	250·5	711·7	219·47	863·53	75·34	1158·34	841	13·46	0·07430
32·0	254·0	715·2	223·02	860·74	75·65	1159·41	791	12·67	0·07891
34·0	257·4	718·6	226·47	858·03	75·95	1160·45	748	11·97	0·08355
36·0	260·7	721·9	229·82	855·41	76·23	1161·46	709	11·35	0·08811
38·0	263·9	725·1	233·07	852·86	76·50	1162·43	674	10·79	0·09269
40·0	267·0	728·2	236·22	850·41	76·75	1163·38	640	10·28	0·09729
42·0	270·0	731·2	239·27	848·01	77·01	1164·29	614	9·8298	0·10173
44·0	272·9	734·1	242·22	845·71	77·25	1165·18	588	9·4123	0·10623
46·0	275·7	736·9	245·07	843·48	77·48	1166·03	564	9·0298	0·11074

TABLE No. XXII.—*continued.*

1	2		3	4	5	6	7	8	9
Total Pressure. Pounds per square inch.	Temperature or Boiling Point in Degrees Fahr.		Total Heat required to raise 1 lb. of Water from 32° Fahr. to Boiling Point.	Heat absorbed per pound of Steam in overcoming *Internal* Resistance to Vaporisation.	Heat absorbed per pound of Steam in overcoming *External* Resistance to Expansion.	Total Heat required to generate 1 lb. of Steam from Water at 32° under Constant Pressure. (3+4+5).	Volume of Steam compared with Water at 32°; or cubic feet of Steam generated by 1 cubic foot of Water at 32°.	Volume. Cubic feet per pound.	Weight. Pounds per cubic foot.
Abs.	Fahr.	Abs.	Br. Th. U.	Br. Th. U.	Br. Th. U.	Br. Th. U.			
48·0	278·4	739·6	247·82	841·32	77·71	1166·85	542	8·6785	0·11522
50·0	281·0	742·2	250·47	839·25	77·92	1167·64	522	8·3545	0·11970
52·0	283·5	744·7	253·02	837·26	78·13	1168·41	503	8·0547	0·12417
54·0	285·9	747·1	255·47	835·34	78·33	1169·14	485	7·7764	0·12860
56·0	288·2	749·4	257·82	833·50	78·52	1169·84	469	7·5176	0·13302
58·0	290·4	751·6	260·07	831 60	78·84	1170·51	454	7·2758	0·13746
60·0	292·6	753·8	262·32	829·96	78·90	1171·18	440	7 0497	0·14186
62·0	294·7	755·9	264·47	828·27	79·08	1171·82	427	6·8378	0·14625
64·0	296·8	758·0	266·62	826·59	79·25	1172·46	414	6·6386	0·15062
66·0	298·8	760·0	268·67	824·98	79·42	1173·07	403	6·4511	0·15502
68·0	300·8	762·0	270·72	823·38	79·58	1173·68	392	6·2743	0·15937
70·0	302·8	764·0	272·76	821·79	79·74	1174·29	381	6·1071	0·16376
72·0	304·7	765·9	274·70	820·66	79·51	1174·87	371	5 9180	0·16812
74·0	306·6	767·8	276·64	818·78	80·04	1175·46	362	5·7990	0·17242
76·0	308·5	769·7	278·58	817·26	80·19	1176·03	353		
78·0	310·3	771·5	280·42	815·84	80·32	1176·58			
80·0	312·1	773·3	282·26	814·40	80·47				

88·0									
90·0	320·3	781·5	290·60						
92·0	321·8	783·0	292·20	806·67	81·22				
94·0	323·3	784·5	293·74	805·47	81·33	1180·54			
96·0	324·8	786·0	295·28	804·28	81·44	1181·00	284	4·5482	
98·0	326·3	787·5	296·82	803·09	81·55	1181·46	278	4·4613	0·22416
100	327·7	788·9	298·26	801·96	81·66	1181·88	273	4·3777	0·22843
105	331·2	792·4	301·85	799·21	81·89	1182·95	261	4·1818	0·23911
110	334·6	795·8	305·34	796·52	82·13	1183·99	250	4·0025	0·24987
115	337·9	799·1	308·73	793·92	82·35	1185·00	239	3·8390	0·26050
120	341·1	802·3	312·01	791·43	82·56	1186 00	230	3·6883	0·27114
125	344·2	805·4	315·20	788·97	82·75	1186·92	221	3·5491	0·28177
130	347·2	808·4	318·29	786·62	82·94	1187·85	213	3·4203	0·29240
135	350·1	811·3	321·28	784·32	83·12	1188·72	206	3·3008	0·30295
140	352·9	814·1	324·16	782·12	83·29	1189·57	199	3·1896	0·31351
145	355·6	816·8	326·94	780·00	83·46	1190·40	192	3·0859	0·32410
150	358·3	819·5	329·72	777·87	83·63	1191·22	186	2·9888	0·33460
160	363·4	824·6	334·97	773·86	83·94	1192·77	175	2·8126	0·35556
170	368·3	829·5	340·02	770·00	84·25	1194·27	166	2·6570	0·37640
180	373·0	834·2	344·87	766·27	84·56	1195·70	157	2·5184	0·29706
190	377·5	838·7	349·52	762·69	84·86	1197·07	149	2·3943	0·21766
200	381·8	843·0	353·97	759·27	85·15	1198·39	142	2·2824	0·43811
210	385·8	847·0	358·12	756·05	85·43	1199·60	136	2·1810	0·45851
220	389·7	850·9	362·17	752·94	85·69	1200·80	130	2·0881	0·47890
230	393·6	854·8	366·22	749·83	85·94	1201·99	125	2·0031	0·49920
240	397·3	858·5	370·07	746·86	86·18	1203·11	120	1·9251	0·51943
250	400·8	862·0	373·72	744·04	86·42	1204·18	115	1·8533	0·53986
275	409·1	870·3	382·37	737·34	87·01	1206·72	106	1·6962	0·58955
300	417·1	878·3	390·72	730·85	87·59	1209·16	97	1·5653	0·63890

TABLE No. XXII.—*continued.*

1	2		3	4	5	6	7	8	9
Total Pressure. Pounds per square inch.	Temperature or Boiling Point in Degrees Fahr.		Total Heat required to raise 1 lb. of Water from 32° Fahr. to Boiling Point.	Heat absorbed per pound of Steam in overcoming *Internal* Resistance to Vaporisation.	Heat absorbed per pound of Steam in overcoming *External* Resistance to Expansion.	Total Heat required to generate 1 lb. of Steam from Water at 32° under Constant Pressure. (3+4+5).	Volume of Steam compared with Water at 32°; or cubic feet of Steam generated by 1 cubic foot of Water at 32°.	Volume. Cubic feet per pound.	Weight. Pounds per cubic foot.
Abs.	Fahr.	Abs.	Br. Th. U.	Br. Th. U.	Br. Th. U.	Br. Th. U.			
48·0	278·4	739·6	247·82	841·32	77·71	1166·85	542	8·6785	0·11522
50·0	281·0	742·2	250·47	839·25	77·92	1167·64	522	8·3545	0·11970
52·0	283·5	744·7	253·02	837·26	78·13	1168·41	503	8·0547	0·12417
54·0	285·9	747·1	255·47	835·34	78·33	1169·14	485	7·7764	0·12860
56·0	288·2	749·4	257·82	833·50	78·52	1169·84	469	7·5176	0·13302
58·0	290·4	751·6	260·07	831 60	78·84	1170·51	454	7·2758	0·13746
60·0	292·6	753·8	262·32	829·96	78·90	1171·18	440	7 0497	0·14186
62·0	294·7	755·9	264·47	828·27	79·08	1171·82	427	6·8378	0·14625
64·0	296·8	758·0	266·62	826·59	79·25	1172·46	414	6·6386	0·15062
66·0	298·8	760·0	268·67	824·98	79·42	1173·07	403	6·4511	0·15502
68·0	300·8	762·0	270·72	823·38	79·58	1173·68	392	6·2743	0·15937
70·0	302·8	764·0	272·76	821·79	79·74	1174·29	381	6·1071	0·16376
72·0	304·7	765·9	274·70	820·66	79·51	1174·87	371	5 9480	0·16812
74·0	306·6	767·8	276·64	818·78	80·04	1175·46	362	5·7990	0·17245
76·0	308·5	769·7	278·58	817·26	80·19	1176·03	353	5·6568	0·17680
78·0	310·3	771·5	280·42	815·84	80·32	1176·58	345	5·5214	0·18110
80·0	312·1	773·3	282·26	814·40	80·47	1177·13	337	5·3926	0·18545
82·0	313·8	775·0	284·00	813·05	80·60	1177·65	329	5·2698	0·18977
84·0	315·5	776·7	285·74	811·70	80·73	1178·17	320	5·1526	0·19410
86·0	317·1	778·3	287·38	810·41	80·86	1178·65	315	5·0406	0·19840

88·0	318·7	779·9	289·02	809·15	80·98	1179·15	308	4·9336	0·20260
90·0	320·3	781·5	290·66	807·86	81·11	1179·63	301	4·8311	0·20700
92·0	321·8	783·0	292·20	806·67	81·22	1180·09	295	4·7329	0·21130
94·0	323·3	784·5	293·74	805·47	81·33	1180·54	289	4·6387	0·21560
96·0	324·8	786·0	295·28	804·28	81·44	1181·00	284	4·5482	0·21995
98·0	326·3	787·5	296·82	803·09	81·55	1181·46	278	4·4613	0·22416
100	327·7	788·9	298·26	801·96	81·66	1181·88	273	4·3777	0·22843
105	331·2	792·4	301·85	799·21	81·89	1182·95	261	4·1818	0·23911
110	334·6	795·8	305·34	796·52	82·13	1183·99	250	4·0025	0·24987
115	337·9	799·1	308·73	793·92	82·35	1185·00	239	3·8390	0·26050
120	341·1	802·3	312·01	791·43	82·56	1186 00	230	3·6883	0·27114
125	344·2	805·4	315·20	788·97	82·75	1186·92	221	3·5491	0·28177
130	347·2	808·4	318·29	786·62	82·94	1187·85	213	3·4203	0·29240
135	350·1	811·3	321·28	784·32	83·12	1188·72	206	3·3008	0·30295
140	352·9	814·1	324·16	782·12	83·29	1189·57	199	3·1896	0·31351
145	355·6	816·8	326·94	780·00	83·46	1190·40	192	3·0859	0·32410
150	358·3	819·5	329·72	777·87	83·63	1191·22	186	2·9888	0·33460
160	363·4	824·6	334·97	773·86	83·94	1192·77	175	2·8126	0·35556
170	368·3	829·5	340·02	770·00	84·25	1194·27	166	2·6570	0·37640
180	373·0	834·2	344·87	766·27	84·56	1195·70	157	2·5184	0·29706
190	377·5	838·7	349·52	762·69	84·86	1197·07	149	2·3943	0·21766
200	381·8	843·0	353·97	759·27	85·15	1198·39	142	2·2824	0·43811
210	385·8	847·0	358·12	756·05	85·43	1199·60	136	2·1810	0·45851
220	389·7	850·9	362·17	752·94	85·69	1200·80	130	2·0881	0·47890
230	393·6	854·8	366·22	749·83	85·94	1201·99	125	2·0031	0·49920
240	397·3	858·5	370·07	746·86	86·18	1203·11	120	1·9251	0·51943
250	400·8	862·0	373·72	744·04	86·42	1204·18	115	1·8533	0·53986
275	409·1	870·3	382·37	737·34	87·01	1206·72	106	1·6962	0·58955
300	417·1	878·3	390·72	730·85	87·59	1209·16	97	1·5653	0·63890

TABLE No. XXIII.

Expansion of Rigid Bodies by Heat.

Material.	Expansion between 32° and 212° F. per Unit of Length.	Material.	Expansion between 32° and 212° F. per Unit of Length.
Lead	0·0029	Wrought iron	0·0012
Bronze	0·0018	Steel	0·0011 to 0·0012
Oak	0·0008		
Cast iron	0·0011	Pine	0·0035
Copper	0·0016	Zinc	0·003

TABLE No. XXIV.

Melting Points of various Materials.

Material.	Degrees Fahr.
Antimony	800
Lead	630
Bronze, average	1650
Gold	2190
Cast iron, white	1930
Do. grey	2190
Copper	2010
Platinum	4530
Quicksilver	−40
Wrought iron	2730 to 3230
Sulphur	230
Silver	1830
Steel	2430
White metal	410 to 500
Bismuth	500
Bismuth, 8 parts; tin, 3 parts; lead, 5 parts	210
Zinc	800
Tin	450

TABLE No. XXVA.

Specific Gravity of Woods.

Material.	Specific Gravity.	
	Green.	Seasoned.
Oak	0·89 to 1·06	0·65 to 0·92
Ash	0·78 „ 0·93	0·54 „ 0·85
Pine	0·79 „ 0·99	0·38 „ 0·48
Fir	0·81 „ 1·00	0·46 „ 0·76
Cork	—	0·24
Pitch pine	—	0·66
Guaiacum	—	1·26 to 1·34
Teak	—	0·88 „ 0·98

TABLE No. XXVB.

Specific Gravity of Metals.

Material.	Specific Gravity.	Material.	Specific Gravity.
Aluminium	2·6 to 2·7	Iron—	
Antimony	6·7	Pig, white steel	7·6
Lead	11·3 to 11·4	Siemens-Martin mild steel	7·85
Bronze—		Siemens-Martin steel	7·85
Delta metal	8·6	Tool steel	7·86
Bell metal	8·8	Copper	8·8 to 9
Manganese bronze	8·5	German silver	8·4 „ 8·7
Brass	8·4 to 8·7	Platinum	21·5
Muntz metal	8·5	Quicksilver	13·6
Phosphor bronze	8·8	Silver	10·5
Gunmetal	8·7	White metal	7·1
Iron—		Tin	7·3
Pure	7·8	Zinc	7·1
Cast	7·25		
Pig, grey	6·8 to 7·5		

TABLE No. XXVc.

Specific Gravity of Various Materials.

Material.	Specific Gravity.	Material.	Specific Gravity.
Asbestos - -	1·2	Chalk - - -	1·8 to 2·6
Asphalt - -	1·1 to 1·5	Marble - - -	2·7
Cement, powder -	1·15 „ 1·7	Cotton wool, fairly loose - -	0·16
„ set - -	2·7 „ 3	Sulphur - -	2·0
Hard fat - -	0·92 „ 0·94	Talc or Mica -	0·91
Rubber, raw - -	0·92 „ 0·96	Bricks, ordinary -	1·94
„ vulcanised	1·45	„ fire - -	1·91

TABLE No. XXVd.

Relative Weights of Coals per Unit Volume (Water = 1).

Material.	Relative Weight (Water = 1).	Material.	Relative Weight (Water = 1).
Anthracite - -	1·4 to 1·7	Coke, screened -	0·4
German brown coal	0·8 „ 1·5	Hard coal - -	1·2 to 1·5
Coke - - -	1·4	„ screened	0·75

TABLE No. XXVe.

Specific Gravity of Fluids.

Material.	Specific Gravity.	Material.	Specific Gravity.
Alcohol, pure -	0·79	Hydrochloric acid (concentrated) -	1·19
Turpentine - -	0·85	Sulphuric acid (concentrated) - -	1·84
Linseed oil - -	0·94	Sea-water - -	1·025
Mineral oil - -	0·90 to 0·92	Distilled water at 4° Centigrade or 39° Fahr. - -	1·00
Rape oil - -	0·91		
Petroleum - -	0·80 to 0·90		
Quicksilver - -	13·6		
Nitric acid (concentrated) - -	1·53		

TABLE No. XXVF.

Specific Weights of Gases at 30 *inches of Mercury and* 32° *Fahr.* (*Water* = 1).

Gas	Specific Weight.	Gas.	Specific Weight.
Carbonic oxide -	0·001250	Steam (see Table XXII.).	
Carbonic acid gas -	0·001978		
Coal gas - -	0·000690	Oxygen - -	0·001429
Air - - -	0·001293	Nitrogen - -	0·001256
Methane, marsh gas	0·000720	Hydrogen - -	0·0000896

If G is the specific weight of a gas at 32° Fahr. and 30 inches, and G_1 the specific weight of the same gas at t° Fahr. and p inches of mercury—

$$G_1 = G \frac{p}{30}\left(\frac{1}{1+\alpha t}\right)$$

in which the coefficient α is nearly the same for all gases and = ·002036 per degree Fahr.

735 mm. of mercury
28·6 inches of mercury } = 1 atmosphere = { 1 kg. per square cm.
14·22 lb. per square inch.

TABLE No. XXVI.

Strength and Elasticity of Various Materials.

Material.	Breaking Strength. Tons per square inch.*	Elongation per cent. on an 8 inch Length.	Elastic Limit. Tons per square inch.	Modulus of Elasticity. Tons per square inch.
Common grey cast iron	7·6 to 9	—	—	—
Common grey cast iron under compression	44 „ 47	—	—	9,530
Extra good cast iron, for cylinders and the like	10·5 „ 13·75	—	—	—
Extra good cast iron, for cylinders and the like, under compression	47 to 51	—	—	6,650
Good welding iron, small forgings	22 „ 24	14 „ 18	12 to 16	12,700
Siemens-Martin mild forged steel, according to percentage of carbon present	24 „ 27	20 „ 25	12 „ 19	13,600
Siemens-Martin forged steel, for shafts, &c.	29 „ 35	20 „ 25	12 „ 22	14,000
Forged crucible steel, best quality	29 „ 35	20 „ 25	12 „ 22	14,000
Nickel steel, forged	35 „ 41	20	24	14,000
Tool steel, unhardened	48 „ 57	—	35 and over	14,000
Steel castings (Siemens-Martin)	25 „ 32	18 to 20	12 to 19	13,600
Boiler plates of mild steel	24 „ 27	20 „ 25	16	13,600
Best crucible steel wire for ropes	95 „ 115	—	—	—
Rolled brass	9·5	—	—	7,000

* For strength in compression or shear, at the most four-fifths of the strength in tension may be assumed.

TABLE No. XXVI.—*continued.*

Material.	Breaking Strength. Tons per square inch.*	Elongation per cent. on an 8 inch Length.	Elastic Limit. Tons per square inch.	Modulus of Elasticity. Tons per square inch.
Best gunmetal, bronze for valves, &c.	12 to 19	at least 10 to 20	—	5,700
Muntz metal, rolled or forged	22	—	—	—
Delta metal, forged	22 to 24	—	11·5	6,350
Manganese bronze,† cast (propeller blades)	19 „ 29	15 to 25	—	—
Manganese bronze, drawn (bolts)	25 „ 32	20 „ 40	—	—
Copper plates	13 „ 15	38	9	7,000
Copper wire	Up to 24	—	—	—
Oak (with the grain)	7 tension 4·2 compression	—	—	760
Pine (with the grain)	7 tension 2·8 compression	—	—	760
Ash (with the grain)	7·6 tension 4·2 compression	—	—	630

Note.—Unless otherwise mentioned, the table refers to the breaking strength in tension.

The figures for elongation give the percentage increase of the original length which occurs in a test bar 8 inches long when broken.

* For strength in compression or shear, at the most four-fifths of the strength in tension may be assumed.

† Compare Table No. XXVII.

TABLE No. XXVII.

*Strength and Elasticity of Manganese Bronze.** (*Parsons' Manganese Bronze.*)

QUALITY No. 1.—This cannot be cast, but is only obtainable hammered, rolled, or drawn.

Uses.—Screws, spindles, nuts, piston rods, circulating pump shafts, &c.

Strength.—22 to 39 tons per square inch.

Elastic Limit.—11 to 16 tons per square inch (and up to 34 tons per square inch).

Extension.—20 to 45 per cent.

QUALITY No. 2.—For castings.

Uses.—Propeller blades, sundry engine parts in torpedo-boats, stern posts for small vessels, &c.

Strength.—23·5 to 37 tons per square inch.

Elastic Limit.—15 to 20·5 tons per square inch.

Extension.—18 to 29 per cent.

Note.—This extension is that obtained on a test piece 2 inches long. The highest values given for strength and extension are those of a single test.

* From data given by the firm.

TABLE No. XXX.

Moments of Inertia "I" and Internal Moments of Resistance or Moduli of Section "W" for Circular Sections of Diameter "d."

d	$I = \frac{d^4\pi}{64}$	$W = \frac{d^3\pi}{32}$	d	$I = \frac{d^4\pi}{64}$	$W = \frac{d^3\pi}{32}$
10	490·9	98·17	26	22,432	1,726
10·5	596·4	113·6	27	26,087	1,932
11	718·7	130·7	28	30,172	2,165
11·5	858·5	149·3	29	34,719	2,394
12	1,018	169·6	30	39,761	2,651
12·5	1,179	188·6			
13	1,402	215·7	31	45,333	2,925
13·5	1,630	241·5	32	51,472	3,217
14	1,886	269·4	33	58,214	3,528
14·5	2,178	299·3	34	65,597	3,859
15	2,485	331·3	35	73,662	4,209
			36	82,448	4,580
15·5	2,833	365·6	37	91,998	4,973
16	3,217	402·1	38	102,354	5,387
16·5	3,638	441·0	39	113,561	5,824
17	4,100	482·3	40	125,664	6,283
17·5	4,604	526·2			
18	5,153	572·6	41	138,709	6,766
18·5	5,750	621·6	42	152,745	7,274
19	6,397	673·4	43	167,820	7,806
19·5	7,098	728·0	44	183,984	8,363
20	7,854	785·4	45	201,289	8,946
21	9,547	909·2	46	219,787	9,556
22	11,499	1,045	47	239,531	10,193
23	13,737	1,194	48	260,576	10,857
24	16,286	1,357	49	282,979	11,550
25	19,175	1,534	50	306,796	12,272

TABLE No. XXX.—*continued.*

d	$I = \frac{d^4\pi}{64}$	$W = \frac{d^3\pi}{32}$	d	$I = \frac{d^4\pi}{64}$	$W = \frac{d^3\pi}{32}$
51	332,086	13,023	76	1,637,662	43,096
52	358,908	13,804	77	1,725,571	44,820
53	387,323	14,616	78	1,816,972	46,589
54	417,393	15,459	79	1,911,967	48,404
55	449,180	16,334	80	2,010,619	50,265
56	482,750	17,241			
57	518,166	18,181	81	2,113,051	52,174
58	555,497	19,155	82	2,219,347	54,130
59	594,810	20,163	83	2,329,605	56,135
60	636,172	21,206	84	2,443,920	58,189
61	679,651	22,284	85	2,562,392	60,292
62	725,332	23,398	86	2,685,120	62,445
63	773,272	24,548	87	2,812,205	64,648
64	823,550	25,736	88	2,943,748	66,903
65	876,240	26,961	89	3,079,853	69,210
66	931,420	28,225	90	3,220,623	71,569
67	989,166	29,527	91	3,366,165	73,982
68	1,049,556	30,869	92	3,516,586	76,448
69	1,112,660	32,251	93	3,671,992	78,968
70	1,178,588	33,674	94	3,832,492	81,542
71	1,247,393	35,138	95	3,998,198	84,173
72	1,319,167	36,644	96	4,169,220	86,859
73	1,393,995	38,192	97	4,345,671	89,601
74	1,471,963	39,783	98	4,527,664	92,401
75	1,553,156	41,417	99	4,715,315	95,259
			100	4,908,738	98,175

TABLE No. XXXI.

Bending Moments.

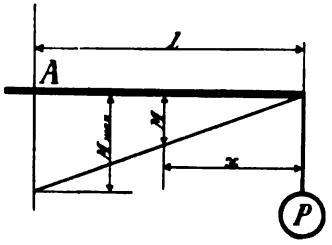
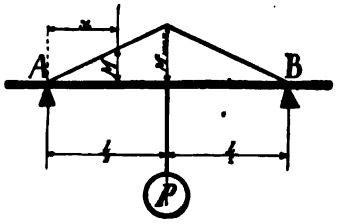
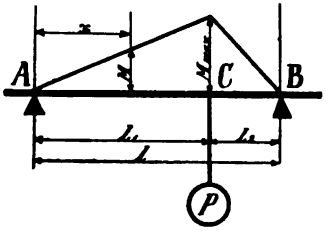
Method of Support. Curve of Moments.	Pressure on Supports A and B. Bending Moment M and Maximum Bending Moment M_{max}.
	$A = P$ $M = P \cdot x$ $M_{max} = P \cdot l$ Cantilever loaded at end. Weakest section at A.
	$A = B = \frac{P}{2}$ $M = \frac{P}{2} x$ $M_{max} = \frac{P \cdot l}{4}$ Beam supported at ends. Weakest section in middle.
	$A = P \cdot \frac{l_2}{l}$ $B = P \cdot \frac{l_1}{l}$ $M = P \cdot \frac{l_2}{l} \cdot x$ $M_{max} = \frac{P \cdot l_2 \cdot l_1}{l}$ Beam unevenly loaded. Weakest section at C.

TABLE No. XXXI.—*continued.*

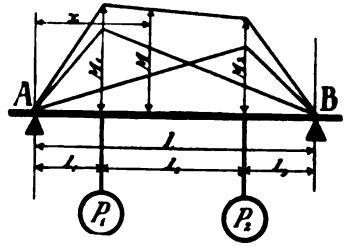
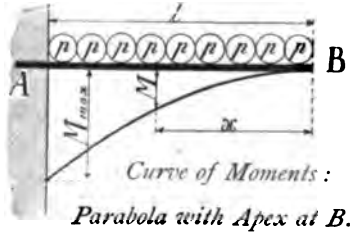
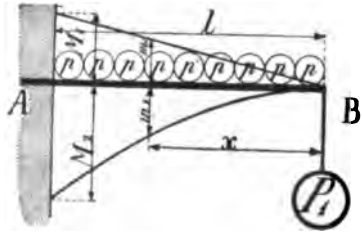
Method of Support. Curve of Moments.	Pressures on Supports A and B. Bending Moment M and Maximum Bending Moment M_{max}
	$A = \frac{P_1(l_2 + l_3) + P_2 l_3}{l}$ $B = \frac{P_2(l_1 + l_2) + P_1 l_1}{l}$ $M_1 = Al_1 \quad M_2 = Bl_3$ $M = A \cdot x - P_1(x - l_1).$ $= B(l - x) - P_2(l - l_3 - x).$ Beam with two loads and unfixed ends. Weakest section at P_1.
 *Curve of Moments: Parabola with Apex at B.*	$A = \Sigma p = P = p \cdot l$ $M = \frac{px^2}{2}$ $M_{max} = \frac{p \cdot l^2}{2} = \frac{P \cdot l}{2}$ p = load per unit length. Cantilever. Evenly distributed load. Weakest section at A.
	$A = P_1 + P = P_1 + pl$ $M = m_1 + m_2 = P_1 x + \frac{px^2}{2}$ $M_{max} = M_1 + M_2 = P_1 l + \frac{pl^2}{2}$ Cantilever. Combined load, evenly distributed and at end. Weakest section at A.

TABLE No. XXXI.—*continued.*

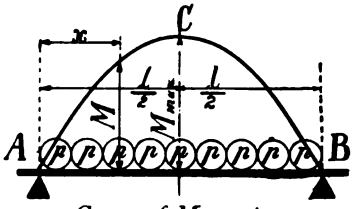
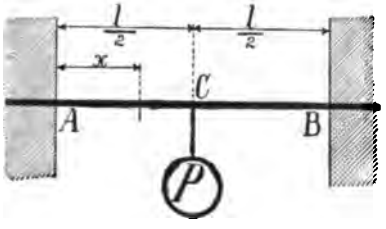
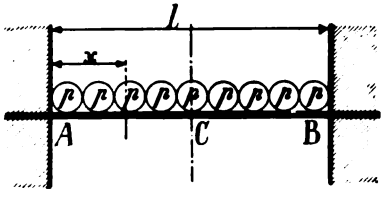
Method of Support. Curve of Moments.	Pressures on Supports A and B. Bending Moment M, and Maximum Bending Moment M_{max}.
 *Curve of Moments: Parabola with Apex at C.*	$A = B = \frac{pl}{2} = \frac{P}{2}$ $M = p\frac{x}{2}(l - x)$ $M_{max} = \frac{p \cdot l^2}{8} = \frac{Pl}{8}$ Weakest section at C. Beam with evenly distributed load and unfixed ends.
	$A = B = \frac{P}{2}$ $M = \frac{Pl}{2}\left(\frac{x}{l} - \frac{1}{4}\right)$ $M_{max} = \frac{P \cdot l}{8}$ in A, B, and C. Weakest sections at A, B, and C. Built-in beam centrally loaded.
	$A = B = \frac{p \cdot l}{2} = \frac{P}{2}$ $M = \frac{pl^2}{2}\left(\frac{1}{6} - \frac{x}{l} + \frac{x^2}{l^2}\right)$ $M_{max} = \frac{Pl}{12}$; $M_c = \frac{Pl}{24}$ Weakest sections at A and B. Built-in beam with evenly distributed load.

TABLE No. XXXI.—*continued.*

Bending Moments.

Let M = bending moment in inch-pounds at any section.

I = moment of inertia of the section taken round that neutral axis which, lying in the plane of the section, passes through its centre of gravity, and is normal to the direction of the bending force.

e = the distance from the neutral axis of the most highly stressed fibre.

$W = \frac{I}{e}$ = internal moment of resistance (or modulus) of the section in inches^3.

Then the permissible bending or skin stress:—

$$s = \frac{M}{W} \text{ lb. per square inch.}$$

For s there should be inserted the permissible values of stress when under tension.

TABLE No. XXXII.

Torsional Strength.

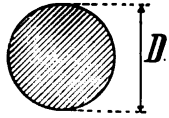
M = twisting moment in inch-pounds.

G = torsional modulus of elasticity = about 5,000 tons per square inch for iron and steel.

l = length in inches of specimen twisted.

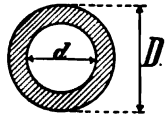
S = torsional stress about four-fifths of permissible tensile stress.

1. Circular Section.



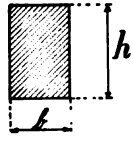
$$M = \frac{\pi}{16} D^3 . S$$

2. Ring Section.



$$M = \frac{\pi}{16} \frac{D^4 - d^4}{D} . S$$

3. Rectangular Section.



$$M = \frac{2}{9} b^2 h . S. \quad (h > b.)$$

4. If ϕ is arc through which shaft is twisted, then for round or circular section—

$$\phi = \frac{32}{\pi D^4} . \frac{M}{G} l$$

ϕ is the length of the arc, measured on a radius of 1 inch, through which two sections at a distance l from each other are twisted.

Table No. XXXIII.

Strength of Struts.

l = length of strut in inches.

I = smallest moment of inertia in inches^4.

E = modulus of elasticity pounds per square inch.

P = load in pounds at which bending first occurs.

d = diameter of strut in inches.

Case 1. One end of strut built in, the other end free.

$$P = \frac{\pi^2}{4} \frac{EI}{l^2} \qquad \text{Fig. I.}$$

Case 2. Both ends free, but constrained to move in the direction of load.

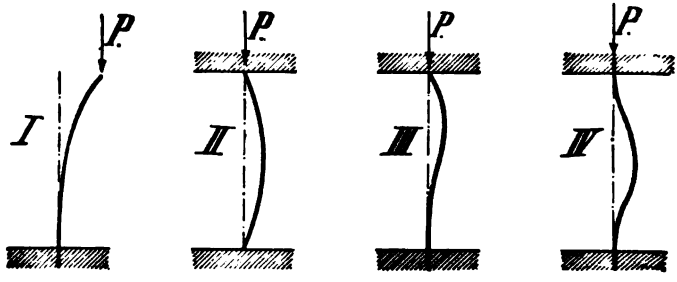
$$P = \frac{\pi^2 EI}{l^2} \qquad \text{Fig. II.}$$

Case 3. One end built in, the other free but constrained to move in direction of load.

$$P = \frac{2\pi^2 EI}{l^2} \qquad \text{Fig. III.}$$

Case 4. Both ends built in.

$$P = 4\pi^2 \frac{EI}{l^2} \qquad \text{Fig. IV.}$$



The above formulæ apply for struts with the proportion—

Case 1. $l = 5d$ for cast iron ; $l = 12d$ for wrought iron.

„ 2. $l = 10d$ „ ; $l = 24d$ „

„ 3. $l = 14d$ „ ; $l = 33d$ „

„ 4. $l = 20d$ „ ; $l = 48d$ „

TABLE No. XXXIV.

German Lloyd's Rules for Iron and Steel for Boilers.

1. *Wrought-iron Plates.*

	Shell Plates.			End Plates.			Furnace Plates.		
	Tensile Strength.		Per cent. Elongation in 8 inches	Tensile Strength.		Per cent. Elongation in 8 inches.	Tensile Strength.		Per cent. Elongation in 8 inches
	tons per sq. in.	kg. per sq. mm.		tons per sq. in.	kg. per sq. mm.		tons per sq. in.	kg. per sq. mm.	
With the grain	21	33	7	22	35	12	23	36	18
Across the grain	19	30	5	21	33	8	21·5	34	12

For plates more than 1 inch thick the strength may be reduced by 0·32 ton per square inch (0·5 kilogramme per square millimetre) for every extra 0·08 inch (2 millimetres) of thickness.

2. *Steel and Mild Steel Plates.*

Tensile strength	Tons per sq. in.	30·5	30	29	28·5	28	27·5	27
	Kg. per sq. mm.	48	47	46	45	44	43	42
Per cent. elongation in 8 inches		20	20	20·5	21	21·5	22	22·5
Tensile strength	Tons per sq. in.	26	25·5	25	24	23·5	23	22
	Kg. per sq. mm.	41	40	39	38	37	36	35
Per cent. elongation in 8 inches		23	23·5	24	24·5	25	25·5	26

For shell plates 20 per cent. elongation is sufficient. Furnace plates and end plates should have a tensile strength not exceeding 28 tons per square inch, with a corresponding elongation.

TABLE No. XXXIV.—*continued.*

3. *Stays and Rivets of Iron or Steel.*

	Wrought Iron.			Steel.		
	Tensile Strength.		Per cent. Elongation in 8 inches.	Tensile Strength.		Per cent. Elongation in 8 inches.
	tons per sq. in.	kg. per sq. mm.		tons per sq. inch.	kg. per sq. mm.	
Stays -	—	—	—	22 to 29	35 to 45	—
Rivets -	at least 23	at least 36	at least 15	22 „ 29	35 „ 45	—
Angles -	at least 23	at least 36	at least 16	—	—	—

TABLE No. XXXV.

German Lloyd's Rules for Steel and Cast Steel for Parts of Engines.

Crank Shafts, Propeller Shafts, and Lengths of Shafting in Wrought Steel.

Strength, 25 to 30 tons per square inch; elongation, 20 per cent. on 8 inches.

Special and nickel steel to have at least 20 per cent. elongation.

Stem and Stern Posts, &c., in Cast Steel.

Strength, 25 to 35 tons per square inch; elongation, 15 per cent. on an 8 inch length.

Note.—For the cast steel portions of engines and boilers 25 to 28 tons is now usually allowed.

TABLE No. XL.

Weight of Machinery (from various published data).

Under heading "Engines" are included main engines with propeller and shafting.

Under heading "Boilers" are included boilers with small and large mountings, fire box, and funnel (boiler empty).

Under heading "Pipework" are included all pumps, water, steam, and exhaust pipes, gratings, platforms, &c.

Type of Engine.	Weight in pounds per H.P.			
	Engines.	Boilers.	Pipework.	Total.
Torpedo-boats and destroyers. Triple-expansion engines*—				
Water-tube boilers	18 to 31	22 to 34	9 to 18	50 to 83
Light cruisers. Triple-expansion engines*—				
With cylindrical boilers	45 ,, 67	78 ,, 100	22 ,, 34	145 ,, 200
With water-tube boilers	45 ,, 67	45 ,, 67	22 ,, 34	113 ,, 124
Battleships. Triple-expansion engines*—				
Cylindrical boilers	67 ,, 83	90 ,, 113	34 ,, 45	190 ,, 204
Fast liners. Cylindrical boilers—				
Triple-expansion engines	113 ,, 135	124 ,, 146	54 ,, 67	290 ,, 350
Quadruple-expansion engines	126 ,, 157	135 ,, 170	54 ,, 67	314 ,, 390
Cargo boats. Cylindrical boilers—				
Large, boats with quadruple-expansion engines	170 ,, 214	170 ,, 214	100 ,, 113	440 ,, 530
Medium-sized, with triple-expansion engines	158 ,, 200	170 ,, 214	90 ,, 100	416 ,, 510
Small, with compound engines	145 ,, 180	170 ,, 190	78 ,, 94	395 ,, 465
Steam pinnaces. Twin or compound engines—				
Locomotive boilers	16 ,, 30	45 ,, 67	9 ,, 18	70 ,, 115

* The weights have been calculated on the maximum H.P. with forced draught.

Table No. XLI.

*Weight of Boiler Equipments Compiled from the German Navy.**

(Weights of boilers without water, but with large and small mountings, fire box, and funnel, as well as all pipework and auxiliary machinery in the boiler-room.)

Type of Boiler.	Weight per H.P. in pounds (reckoned on the maximum continuous H.P.).
Cylindrical boiler - - -	143 to 188, mean 163
Locomotive boiler - - -	71 „ 157, „ 108
Dürr boiler - - - -	89 „ 138, „ 130
Thornycroft boiler - - -	64 „ 92, „ 82
Belleville boiler - - -	100 „ 108, „ 105

This table does not include weights of boilers for torpedo boats.

Table No. XLII.

Weight of Cylindrical Boilers (unclothed and without mountings, &c.).

Cylindrical single-ended boilers, weight per square foot of heating surface—

For 114 to 142 lb. per square inch, 31·2 to 39·5 lb.
For 170 to 214 lb. per square inch, 35·4 to 49 lb.

Double-ended boilers of the same diameter, and of double the heating surface of the corresponding single-ended boilers, weigh about 12 to 17 % per square foot of heating surface less than the latter.

* From Köhn v. Jaski, *Marine-Rundschau*, 1901, v.

APPENDIX.

FINAL REPORT (JUNE 1904) TO THE LORDS COMMISSIONERS OF THE ADMIRALTY OF THE COMMITTEE ON NAVAL BOILERS.

THE COMMITTEE ON NAVAL BOILERS.

H.M.S. "BULWARK" AT RAPALLO,
12th June 1904.

SIR,—I have the honour to submit, herewith, to be laid before the Lords Commissioners of the Admiralty, the final Report of the Boiler Committee of which I am the President. Although I have not been present at the experiments carried out during the last two years, I have received from time to time all the reports, and they show the great care and pains taken by the Committee to obtain correct results.

2. With reference to our previous Report, I am compelled to say that my experience with the Belleville boilers on the Mediterranean Station has been very favourable to them as a steam generator, and it is clear to me that the earlier boilers of this description were badly constructed and badly used. We have had no serious boiler defects in any of the ships out here, and the fact that two ships are about to be recommissioned with only the ordinary annual repairs being undertaken shows that their life is not so short as I originally supposed. However, the second commission of these ships will be a very good test of the capabilities of their boilers.

3. In conclusion, I cannot express too highly my opinion of the work done by my colleagues on the Committee.

I have the honour to be,

Sir,

Your obedient Servant,

(Signed) COMPTON DOMVILE,

Admiral and Commander-in-Chief,
President of the Boiler Committee.

THE SECRETARY TO THE ADMIRALTY.

REPORT OF THE COMMITTEE ON NAVAL BOILERS.

June 1904.

1. The Committee on Naval Boilers appointed by the Lords Commissioners of the Admiralty in September 1900, having completed their investigations and experimental trials, and being in a position to recommend standard types of boiler for use in H.M. Navy, as requested in their Lordships' letter of 28th February 1901, have the honour to submit their final Report.

2. A Statement of the work of the Committee up to May 1902 was given in paragraph 2 of their Report of that date. Since then the reboilering of H.M.S. "Medea" with Yarrow large-tube boilers, and of H.M.S. "Medusa" with Dürr boilers, together with the necessary machinery alterations, have been completed under the supervision of the Committee, and the boilers of both ships have been thoroughly tested. The results obtained are recorded in a separate Report. As requested by their Lordships in their letter S $\frac{16683}{28331}$ of 19th November 1902, the Committee have also carried out a series of trials of the Babcock & Wilcox boilers of H.M.S. "Hermes," which extended from 7th October 1903 to 16th May 1904. These trials also form the subject of a separate Report.

3. The Committee have from time to time reported the results of their investigations, and they have also answered such questions as have been put to them by their Lordships. The Reports and other documents which have already been forwarded include:—

(*a.*) The Interim Report forwarded on the 19th February 1901.

(*b.*) Minutes of the evidence given before the Committee, together with the Appendix thereto, forwarded 26th April 1901.

(*c.*) Report on the trials of the "Hyacinth," "Minerva," and "Saxonia," together with a Summary of Conclusions, forwarded 27th November 1901.

(*d.*) Progress Report for the year 1901, forwarded 31st December, 1901.

(*e.*) Report on the relative economy and efficiency of Belle[illegible] and cylindrical boilers in commissioned ships, forwarded 29th Aug[illegible] 1902.

(*f.*) Report of May 1902, together with the Appendix thereto.

(*g.*) Report on the trials of the "Seagull," "Sheldrake," "Espiègle" and "Fantôme," forwarded 5th August 1902.

There are now submitted with this Report:—

(*h.*) Report on the trials of the "Medea" and "Medusa."

(*i.*) Report on the trials of the "Hermes."

4. The Report of May 1902 was intended to be final as regards the Belleville boiler, and the Committee have since seen no reason to modify the opinion expressed in paragraph 6 of that Report, viz., that it is "undesirable to fit any more of this type in H.M. Navy."

5. In paragraph 5 of their Report of May 1902, the Committee stated that the experience obtained by them since the date of their Report of February 1901, had confirmed them in the opinion that the "advantages of water-tube boilers for Naval purposes are so great, chiefly from a military point of view, that, provided a satisfactory type of water-tube boiler be adopted, it would be more suitable for use in H.M. Navy than the cylindrical type of boiler."

In their Reports of 1901 and 1902, the Committee expressed the opinion that four different types of water-tube boiler, viz.:—

(*a.*) Babcock & Wilcox,
(*b.*) Niclausse,
(*c.*) Dürr, and
(*d.*) Yarrow large-tube,

were sufficiently promising to justify their use in H.M. Navy in combination with cylindrical boilers. Having concluded their experimental investigations, they are now satisfied that two of these four types, viz., the Babcock & Wilcox, similar to that tried in the "Hermes," and the Yarrow large-tube, similiar to that tried in the "Medea," are satisfactory and are suitable for use in battleships and cruisers without cylindrical boilers. In the Babcock & Wilcox boiler, the generating tubes are nearly horizontal; in the Yarrow boiler, they are nearly vertical. Each type has its particular advantages, and only long experience on general service can show which is, on the whole, the better boiler. For the

ıst Dem**ent**, the Committee unanimously recommend both types as suitable
ıval requirements.

of Be- n making these recommendations, the Committee recognise that
d the upkeep of any water-tube boiler is likely to be heavier than that of
cylindrical boiler, but they are of opinion that the two types they
therefore recommend will cost less for upkeep than the other types of large
"Esp**aight-tube** boiler which they have had under trial.

6. The Committee make these recommendations after investigations
ad trials carried out under their superintendence extending over a
eriod of nearly four years.

The ships in which each type of boiler has been tried by the Committee are—

Cylindrical - -	H.M.S. "Minerva" and R.M.S. "Saxonia."
Belleville - -	H.M.S. "Diadem" and H.M.S. "Hyacinth."
Babcock & Wilcox	H.M.S. "Sheldrake," H.M.S. "Espiègle," and H.M.S. "Hermes."
Niclausse - -	H.M.S. "Seagull" and H.M.S. "Fantôme."
Dürr - - -	H.M.S. "Medusa."
Yarrow large-tube	H.M.S. "Medea."

7. Although the Committee have no knowledge of any type of water-tube boiler which is likely to prove more suitable for His Majesty's ships than the two recommended, there are other types which may be considered worthy of trial later on. If any type of boiler is considered, in future, to be of sufficient merit to justify its trial in the Navy, it is recommended that it be fitted in a new vessel not smaller than a second-class Cruiser.

8. As in their previous reports, the Committee do not offer any remarks on the most suitable type of boiler for small vessels of high speed. From the nature of the case, some form of "express" boiler with small tubes closely pitched is absolutely necessary in order to obtain such a ratio of output to weight of boiler as is required in torpedo-boats and destroyers. For small Cruisers, however, which have to keep the sea and act with the fleet, it is probable that a boiler such as the Yarrow large-tube would, on the whole, give better results than the "express" types which have hitherto been fitted.

9. In reference to paragraph 3 (*b*) of their Lordships' letter of 6th September 1900, and to the Committee's Report of May 1902, they desire to call attention to the breakdown of the "Hyacinth's" machinery on 16th February 1903, and to the trouble experienced with the

bearings in the "Hermes" during the homeward run from Gibraltar, which strengthen the recommendation of the Committee contained in paragraph 13 (*a*) of the Report of May 1902, viz., "They consider it desirable, where practicable, to increase the length of stroke and reduce the number of revolutions per minute as compared with the recent practice in His Majesty's Service."

10. The principal comparative results on which the recommendations of the Committee are based are set forth in the succeeding paragraphs. Full details are given in the separate Reports of Trials.

11. *Thermal Efficiency of Boilers.*—The full Tables which are appended to the Committee's Reports give the efficiency of each type of boiler under very varied conditions. The results are here summarised:—

The best obtained with the Babcock & Wilcox boilers of the "Hermes" were during the trials of furnace gas baffling, the boilers in the middle boiler-room, with vertical baffles and a forced air supply over the fires, giving the high efficiency of 81 per cent. on a 30 hours' trial when 20 lbs. of coal were being burnt per square foot of firegrate per hour, and an efficiency of 77·8 per cent. on a 29 hours' trial when burning 27 lbs. per square foot, these rates of combustion corresponding to the ordinary rate of steaming and to the full power of the boilers respectively. The boilers of the "Hermes," with the restricted up-take baffling and without any special air supply over the fires, had a maximum efficiency of 75·8 per cent. on a 12 hours' trial when burning 20·5 lbs. per square foot per hour. On three trials of over 24 hours' duration each and when 19 lbs. were being burnt per square foot per hour, the efficiency was in each case practically 71 per cent. When burning 29 lbs. per square foot per hour for 7 hours, the efficiency of these boilers was 66·3 per cent.; but the weather during this test was so bad that the trial, which was to have been of 8 hours' duration, had to be stopped after the seventh hour on this account. During the baffling trials, however, in good weather, an efficiency of 70·3 was obtained on the 30 hours' trial, when burning 27 lbs. per square foot per hour, or practically the full output.

The maximum efficiency of the Yarrow boilers of the "Medea," viz., 75·7 per cent., was obtained on a 26 hours' trial when burning 18 lbs. per square foot per hour; their efficiency, when burning at the maximum rate of combustion, viz., 40 lbs. per square foot per hour for 8 hours, was 69·5 per cent. On trials of over 24 hours' duration each, burning from 17 to 21 lbs. per square foot per hour, the efficiency remained at or over 75 per cent.

The Belleville boilers of the "Hyacinth" had a maximum efficiency of 77·2 per cent. recorded on a 24½ hours' trial, when 16 lbs. of coal were being burnt per square foot of firegrate per hour. When burning 20 lbs. per square foot per hour for 11 hours, the efficiency was 73·3 per cent.; and burning 17·4 lbs. for 24 hours, it was 71·8 per cent. The efficiency of these boilers on an 8 hours' trial in fine weather, when burning 27 lbs. per square foot per hour, corresponding to the full output of the boiler, was 65 per cent.

The maximum efficiency of the Dürr boilers of the "Medusa" was 64·8 per cent., obtained on an 8 hours' trial, when burning 35 lbs. per square foot per hour, this being the maximum rate of combustion with these boilers; the efficiency, when burning 16 lbs. per square foot per hour for 26 hours, was 63·8 per cent. On trials of over 24 hours' duration each, and burning 18 lbs. and 21 lbs. per square foot, the efficiencies were 61·7 per cent. and 60·3 per cent. respectively.

Of the cylindrical boilers tried, those of the "Saxonia," on the only trial made, which was of 13 hours' duration, and on which 20 lbs. per square foot per hour was burnt, had the high efficiency of 82·3 per cent.

The maximum efficiency obtained with the cylindrical boilers of the "Minerva" was 69·7 per cent., which was recorded on a 25 hours' trial when burning 14 lbs. per square foot per hour; on a trial of 8½ hours' duration, with retarders in the plain tubes and burning 29 lbs. per square foot the efficiency, was 68·4 per cent.

In the smaller ships, the maximum efficiency of the Babcock & Wilcox boilers tried was 66 per cent. on a 12 hours' trial burning 18 lbs. per square foot per hour in the "Sheldrake," and 73·2 per cent. on a 9 hours' trial burning 13 lbs. per square foot in the "Espiègle." The maximum efficiency obtained by the Niclausse boilers of the "Seagull" was 66·9 per cent. on an 8 hours' trial burning 13 lbs. per square foot, and by those of the "Fantôme," 69·8 per cent. on a 9 hours' trial burning 14 lbs. per square foot.

12. A noticeable feature in connection with the boiler efficiencies is the improvement in the results obtained with the later boilers of the Babcock & Wilcox type. The earliest of these fitted in the "Sheldrake" in 1898, showed efficiencies ranging from 66·0 per cent. to 59·2 per cent.; the boilers fitted in the "Espiègle" in 1901 showed improved efficiencies ranging from 73·2 per cent. to 63·1 per cent. Those of the "Hermes," fitted in 1903, show a still further gain in economy, the efficiencies ranging from 75·8 per cent. to 66·3 per cent., and the same boilers, after modification, showed, on one occasion, the high efficiency of 81 per cent. It is noticed in this connection that the three sets of Babcock & Wilcox boilers tried differ from each other in

the arrangement of their heating surface and furnace gas baffling. The boilers of the "Sheldrake" were fitted throughout with tubes $1\frac{15}{16}$ inches in diameter, without any baffles for furnace gases; the boilers of the "Espiègle" were fitted throughout with tubes $3\frac{3}{16}$ inches in diameter, vertical baffles being placed among the tubes and causing a zigzag flow of the gases; the boilers of the "Hermes" were fitted with two rows of $3\frac{15}{16}$ inches diameter tubes immediately over the fire, the remainder of the tubes being $1\frac{13}{16}$ inches diameter, and the baffling of the furnace gases was effected by a restriction of the space for the passage of the gases between the top row of tubes. Those boilers of the "Hermes" which showed the efficiency of 81 per cent. were similar in construction to those last mentioned, but the baffling of the furnace gases was by a vertical system which caused a zigzag flow of the gases over the heating surface, and, in addition, a forced air supply was introduced above the fires (see "Report on the Trials of H.M.S. 'Hermes.'").

The arrangement of the heating surface in both the earlier and later boilers of the Niclausse type was the same, and the thermal efficiencies of the two sets of boilers were very similar.

13. *Wetness of Steam.*—As explained in other Reports, the wetness of the steam was taken throughout the Committee's trials by means of a Carpenter's calorimeter. Experience in the "Medusa" satisfied the Committee that the results registered by this instrument are trustworthy.

As regards the production of *dry* steam at all rates of combustion the Yarrow large-tube and the later Babcock & Wilcox boilers have given the best results.

14. *Loss of Water.*—The loss of feed water with each of the four types of boiler under consideration has been moderate throughout the Committee's trials. In the runs to Gibraltar and back, carried out with the "Medea" and "Medusa," the loss of water was small, being at the rate of 1·6 and 1·8 tons per 1000 horse-power per day respectively. On the 140 hours' endurance trial of the "Hermes," the loss was 3·8 tons per 1000 horse-power per day.

The loss of water may be expected to be greater in boilers fitted with many doors than in those fitted with but few, and to increase as the doors and joints become worn. In this respect, the Yarrow boiler, having only three manhole doors, has an advantage.

15. *Examination and Cleaning of Interiors of Tubes.*—Of the boilers tried by the Committee, the Yarrow boiler can be internally

examined and cleaned in the shortest time and with the least amount of labour—to obtain access for such an examination and cleaning it is only necessary to remove manhole doors. The Babcock & Wilcox type is less easily examined and cleaned—two small doors have to be removed for each tube, and these have to be rejointed after the examination and cleaning have been completed. In order to carry out a thorough examination of the tubes of the Dürr boiler, it is necessary to remove a hand-hole door at the front of each tube, the diaphragm washer of the internal tube, the internal tube itself, and the cap nut at the back end of the generating tube; but, in order to carry out a thorough cleaning, it is also necessary to remove the generator tubes from the boiler, and after the cleaning is complete, these have to be replaced, this being a long and tedious process. The work connected with the examination and cleaning of the tubes of Niclausse boilers is very similar to that necessary with the Dürr boiler. Further, the cap nut at the back end of the Dürr boiler tube permits of each tube being readily emptied, while owing to the back end of Niclausse boilers being inaccessible, some process is necessary to empty the tubes when required, such as blowing the water out of the tube by a special pump and hose.

The necessity for being able to withdraw each of the tubes in a direct line with its axis, renders the clear space required for the installation of Dürr and Niclausse boilers considerably more than would be required for boilers of other types. For warships, where the stokehold space is very limited, this must necessarily cause considerable inconvenience in the arrangement of pipes and auxiliary machinery.

16. *External Cleaning of Tubes.*—In both the "Medea" and the "Hermes" it is possible to partially clean the tubes externally, when the fires are alight, by means of air lances.

The tubes in the "Medea" can be thoroughly cleaned externally when the fires are not alight, as they can be swept in three directions, viz., from the furnace, from the smoke-box, and from the front of the boiler. The tubes of the Dürr and Niclausse boilers cannot be so thoroughly cleaned externally in place as those of the "Medea," the number of rows being greater and the overlapping of the baffles preventing portions of certain tubes being touched. In the Babcock & Wilcox boilers, the tubes can be swept horizontally through side doors fitted to the casings, but, as the boilers in the "Hermes" were originally fitted, the sweeping in a vertical direction was difficult. After the alterations of baffling, the sweeping vertically can be carried out, but necessitates the removal of portions of the baffles. It is to be recognised that any system of baffling among the tubes, however it may

improve the circulation of the gases, renders the cleaning of the tubes themselves more difficult.

17. *Bending of Tubes.*—After the "Medusa" had completed her preliminary runs, it was found that all the tubes of the bottom rows had curved upwards in the middle, the maximum bending being $1\frac{1}{16}$ inches, and these tubes were removed and straightened before starting on the Committee's trials. These tubes had to be straightened again in August 1903, and again at the conclusion of the Committee's trials in February 1904. When the Committee visited H.M.S. "Berwick" in April 1904, it was noticed that the tubes of the bottom rows of the boilers (Niclausse type) were bent upwards, and the members were informed that the maximum bending on the 22nd March 1904 was $\frac{5}{8}$ inch; the ship was new in 1903, and only commissioned in December of that year. With the Niclausse, and also with the Dürr boiler, considerable bending of the tubes at the bottom rows must be expected, and it will be necessary to straighten these tubes when the amount of bending exceeds $\frac{3}{4}$ inch. This will entail a considerable amount of extra work with these types of boiler, which will be off service for corresponding periods. The upward bend of the generator tube is often greater than the space between the inner and outer tubes, and as the inner tube, which is only supported at the two ends, remains straight, it is liable to touch the outer tube at some point, thus impeding the circulation of water between them. To prevent this, it may be necessary to support the inner tube at the middle of its length as well as at the back end, so that it must bend with the outer tube.

In the case of the Yarrow boilers of the "Medea," the Committee experimented in six of the boilers with the firerows of tubes purposely bent, as described on page 9 of the "Report on the Trials of the 'Medea' and 'Medusa,'" with the object of overcoming some slight leakages of tube ends which showed themselves when working under forced draught. In two boilers, the tubes of the firerows were left straight. Although these bent slightly in use, no trouble was experienced with them; and, during the later trials, these boilers proved to be as satisfactory as regards freedom from leakage as those in which the firerows had been put in bent. The Committee have suggested in their letter of the 21st December 1903, concerning the Yarrow boilers proposed for H.M.S. "Warrior," that the tubes of the firerows be bent one inch from the straight, and this recommendation they think should apply to future designs.

In the Babcock & Wilcox boilers of the "Hermes," although some of the tubes of the bottom rows have bent, no leakage of tube ends has

resulted, and it has not been necessary to remove any tubes for straightening or renewal.

18. *Corrosion of Tubes and Wear of Casings and Uptakes.*—In none of the four types of water-tube boiler which were recommended for trial by the Committee has there been any considerable corrosive decay of tubes, and the ordinary wear has been very slight. On the conclusion of the Committee's trials, the tubes of the boilers of the "Medea" and "Hermes" had not deteriorated to any appreciable extent. This applies also to the "Medusa," except that the internal tubes have shown signs of roughening.

In the "Medusa" (Dürr boilers), there was some buckling of the side casings of the boilers, and some of the casing doors at the back of the boilers became warped and burnt.

No trouble was experienced in connection with the casings and uptakes of the Yarrow boilers of the "Medea," and very little with those of the Babcock & Wilcox boilers of the "Hermes." From the experience of the Committee with the boilers of the S.S. "Martello," employed on the Atlantic trade for nearly four years, and also from their experience to date with the "Hermes," it is considered that the durability of the casings and uptakes of Babcock & Wilcox boilers will prove to be satisfactory under the ordinary conditions of Naval Service.

In the Yarrow boiler, the temperature of the furnace gases is considerably reduced before they reach any part of the side casings, and, in consequence of this moderate temperature, the casings and uptakes of the "Medea's" boilers were uninjured on the conclusion of the Committee's trials.

In this respect, the Yarrow boiler is superior to the other types of water-tube boiler which have been tried by the Committee.

19. *Liability to Damage from being Forced.*—The makers of the Dürr boilers stated that not more than 35 lbs. of coal should be burnt per square foot of firegrate per hour in the "Medusa." The Committee consider that this limitation of the quantity of coal to be burnt was prudent, as the overheating and bending of tubes in one of the boilers during the full-power homeward run from Gibraltar were, in the opinion of the Committee, due to the fact that the safe limit had been exceeded. It is also considered that the limitation of the amount of coal to be burnt per square foot of grate applies with even greater force to the Niclausse boiler, as the supply of water to the tubes is freer in the case of the Dürr boiler than in that of the Niclausse. As the result of their trials, the Committee find that the Yarrow boiler can be severely forced

without danger, and that the Babcock & Wilcox boiler can with safety be forced to the extent shown in the Reports.

20. *Skilled Firing Required.*—The satisfactory stoking of water-tube boilers requires a higher degree of skill than that of cylindrical boilers, and this is more necessary with the large grates of the Dürr, Niclausse, and Babcock & Wilcox boilers than with the smaller grates and better shape of combustion chamber of the Yarrow. The stoking in the "Medea," "Medusa," and "Hermes" was good throughout the the trials, and towards the end was excellent. Under ordinary service conditions, such good firing can hardly be expected, at least until a vessel has been some time in commission. Good results can, however, be obtained with Yarrow boilers with engine-room complements new to the ship, as shown by the trials to Malta and back, which have been made by the "Medea" since the completion of the Committee's trials with that vessel.

21. *Superheated Steam.*—The Dürr boiler was the only one tried by the Committee which had any arrangements for superheating. It was fitted with complicated directing plates in the steam collector and with superheater tubes. The fittings in the steam collector are undesirable, and they and the superheater tubes will probably require frequent renewal, while the amount of superheat obtained by their use was small, even when the temperature of the funnel gases was abnormally great. The results obtained were not sufficient to enable the Committee to express any opinion as to the value of superheating as applied to Naval boilers.

22. *Feeding of the Boilers.*—No trouble has been experienced with the feeding of any of the four types of boilers under consideration. In the "Medea" and "Medusa," the boilers were fitted with automatic feed regulators. It was found, however that these were not sufficiently sensitive in opening and closing (allowing a variation of level of about 6 inches in the gauge glass); the feed was therefore regulated throughout the trials by hand, no trouble being experienced in doing this. For a similar reason, the feed was regulated by hand in the "Hermes" during the trials. In the "Medea," the feed regulators were inside the steam collectors and interfered with the examination and cleaning of the middle rows of tubes. The Committee consider that the balance of advantages rests with the omission of automatic feed regulators in boilers such as the Yarrow large-tube and the Babcock & Wilcox, where there is a fairly large reserve of water in the boiler.

23. *Salt Water.*—The Report on the trials of the "Medea" and "Medusa" contains a description of experiments made on the Yarrow and Dürr boilers in regard to their behaviour when working with brackish water. These experiments, so far as they went, indicated that neither type of boiler was likely to give trouble from this cause. In the case of the Yarrow boiler, this result has been corroborated by the fact that on a recent voyage the "Medea" is reported to have had leaky condenser tubes and a corresponding density in the boilers without any bad effect.

24. *Relative Weights.*—In the case of the "Hermes," the "Medea," and the "Medusa," the new boilers were installed without any alterations being made in the stokehold floor spaces. A comparison of weights and maximum output of the boilers gives the following results :—

Type of Boiler.	Ship.	Weight of Boiler-room Installation.	Maximum Output of Steam per hour.	Output of Steam per hour per ton of Boiler-room Weights.	
		Tons.	lbs.	lbs.	
Cylindrical	"Saxonia"	1,000 (abt.)	132,600	132·6	
Ditto -	"Minerva"	567	167,100	295	With retarders.
Ditto -	Ditto	558	156,200	280	As originally fitted.
Belleville -	"Hyacinth"	454	178,700	394	
Yarrow -	"Medea"	330	157,800	478	
Dürr - -	"Medusa"	314	158,000	503	
Babcock & Wilcox -	"Hermes"	490	200,000	410	With vertical baffles, and forced air supply above the fires.
Ditto -	Ditto	481	182,300	380	As originally fitted.
Ditto -	"Sheldrake"	125	43,840	351	
Ditto -	"Espiègle"	95	24,780	261	
Niclausse -	"Seagull"	135	48,450	359	
Ditto -	"Fantôme"	76·5	22,750	297	

25. The Committee are under great obligation to Mr C. J. Wilson, F.C.S., who has, during the four years of their work, given his valuable personal attention to the analysis of funnel gases and of coal samples without any remuneration. They are also much indebted to Messrs Thomas Wilson, Sons, & Company, for permission to examine the boilers of the S.S. "Martello," and to Mr W. S. Hide, the Superintending Engineer of that Company, for affording the Committee facilities for carrying out the inspections and giving information concerning the results obtained in the running of that vessel.

26. The Committee desire in conclusion, to place on record their appreciation of the assistance which they have received from their Secretaries. Captain Browning, R.N., acted as Joint-Secretary until his appointment to H.M.S. "Ariadne" in 1902. Engineer-Lieutenant W. H. Wood, R.N., has continued to act as Secretary throughout their whole work. The diligence and energy which the latter officer has shown in carrying out his work, his knowledge of the scientific as well as of the practical side of marine engineering, and his capacity for dealing both with details and with general organisation, have been invaluable to the Committee throughout, and especially in connection with the carrying out of their boiler trials at sea, a work of no little difficulty and complexity, and they desire to bring his services to the favourable notice of their Lordships.

(Signed) COMPTON DOMVILE,
Admiral and Chairman.

JAS. BAIN.
JOHN INGLIS.
ALEX. B. W. KENNEDY.
JOHN LIST.
J. T. MILTON.
JOS. A. SMITH.

WM. H. WOOD,
Secretary.

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   This Publication includes the Resolutions of the Committee with respect to Standards for Copper Conductors, and contains Tables of Dimensions for Single, Concentric, and Stranded Conductors, Armour, and Di-Electric.

8. British Standard Specification for **Tubular Tramway Poles, 5s.** net.

   This Specification contains the Sizes and Dimensions of Standard Tubular Tramway Poles, together with the Tests recommended by the Committee.

9. British Standard Specification and Sections of **Bull-Headed Railway Rails, 10/6** net.

   This Publication includes the Specification and Sections of Bull-headed Railway Rails. The Sections vary from 60 to 100 lbs. per yard, each Section increasing in weight by 5 lbs. per yard. Fully dimensioned Diagrams of every Section are included, together with Tables of Dimensions, and Sketches showing the forms of Standard Tensile Test Pieces.

10. British Standard Tables of **Pipe Flanges, 2/6** net.

    This Publication contains particulars of Flanges for different ranges of Steam Pressure up to 325 lbs. per square inch, and Water Pressure up to 200 lbs. per square inch. Dimensions of the Standard Bends and Tees recommended by the Committee are also given.

[*Continued on next page.*

Publications of Engineering Standards Committee—*continued.*

11. British Standard Specification and Sections of **Flat-Bottomed Railway Rails, 10/6** net.

This Publication will include the Specification and Sections of Flat-Bottomed Railway Rails. The Sections vary from 20 to 100 lbs. per yard, each Section increasing in weight by 5 lbs. per yard. Fully dimensioned Diagrams of every Section are included, together with Tables of Dimensions, and Sketches showing the forms of Standard Tensile Test Pieces.

12. British Standard Specification for **Portland Cement, 2/6** net.

This Specification contains the Tests for Portland Cement recommended by the Committee. Dimensioned Diagrams are given of the Standard Briquette, Jaws for holding same, Needle for ascertaining setting time, and the Apparatus for conducting the "*Le Chatelier*" Test.

13. British Standard Specification for **Structural Steel for Shipbuilding, 2/6** net.

This Specification includes the Tests recommended by the Committee for Steel Plates, Angles, Channels, Bars, and Rivets used for Shipbuilding, together with Sketches showing the forms of the Standard Tensile Test Pieces.

14. British Standard Specification for **Structural Steel for Marine Boilers, 2/6** net.

This Specification includes the Tests recommended by the Committee for Steel Plates, Angles, Stay Bars, and Rivets used in the Construction of Marine Boilers, together with Sketches showing the forms of the Standard Tensile Test Pieces.

15. British Standard Specification for **Structural Steel for Bridges** (to be published shortly), **2/6** net.

This Specification will include the Tests for Steel used in Bridge Construction, together with Sketches showing the forms of Standard Tensile Test Pieces.

16. British Standard Specification for **Telegraph Material** (to be published shortly), **5**s. net.

This Publication will include the Resolutions of the Committee with respect to Standards for Copper Conductors, and will contain Specifications for general Telegraph Material, together with Tables for Hard and Annealed Copper Wire.

17. Interim Report on **Electrical Machinery** (C.L. 224).

This Report gives Standard Pressures and Frequency; Speeds for Direct Current Generators, D.C. Motors and A.C. Induction Motors. Definitions and Standard Test Conditions are also included; the latter are, however, still under consideration, and must not, therefore, be regarded as final.

18. Forms of **Standard Tensile Test Pieces** (C.L. 116).

This contains Dimensioned Diagrams, and will be inserted in all Specifications in which reference is made to the Standard Tensile Test Pieces.

The above publications can be obtained from CROSBY LOCKWOOD & SON, Publishers to the Committee, 7 Stationers' Hall Court, Ludgate Hill, London, E.C., or at the Offices of the Committee, 28 Victoria Street, Westminster, London, S.W.

NOTE.—With the exception of Nos. 4 and 6, the Volumes are issued in Foolscap size, bound in strong Paper Covers.